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Takahashi et al.

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(54) **HYDRAULIC DRIVE SYSTEM FOR ELECTRICALLY-DRIVEN HYDRAULIC WORK MACHINE**

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(51) **Int. Cl.**
F15B 11/16 (2006.01)
E02F 9/22 (2006.01)
(Continued)

(52) **U.S. Cl.**
CPC ***E02F 9/2242*** (2013.01); ***E02F 9/20*** (2013.01); ***E02F 9/2004*** (2013.01); ***E02F 9/22*** (2013.01);
(Continued)

(58) **Field of Classification Search**
CPC F15B 11/17; E02F 9/2292
See application file for complete search history.

(56) **References Cited**

U.S. PATENT DOCUMENTS

6,989,640 B2 * 1/2006 Yoshimatsu E02F 9/2075 318/34
7,059,125 B2 * 6/2006 Oka E02F 9/2235 60/422

(Continued)

FOREIGN PATENT DOCUMENTS

JP 2003-155760 A 5/2003
JP 2005-83427 A 3/2005

(Continued)

OTHER PUBLICATIONS

International Preliminary Report on Patentability (PCT/IB/338 & PCT/IB/373) issued in PCT Application No. PCT/JP2018/009830 dated Sep. 26, 2019, including English translation of document C2 (Japanese-language Written Opinion (PCT/ISA/237 previously filed on Feb. 21, 2019)) (six (6) pages).

(Continued)

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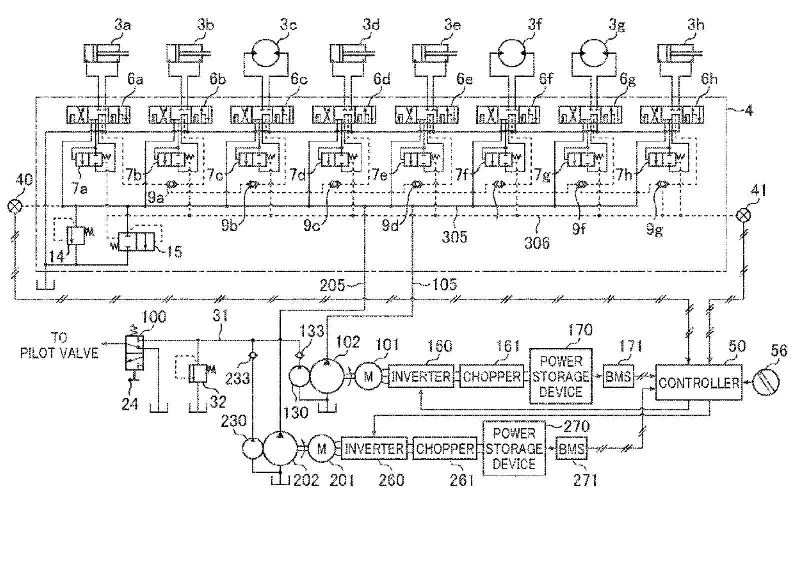
(74) *Attorney, Agent, or Firm* — Crowell & Moring LLP

(57) **ABSTRACT**

A hydraulic drive system for an electrically-driven hydraulic work machine makes it possible to make a rated voltage of various electric equipment such as power storage devices common to one of an electrically-driven hydraulic work machine that is capable of being operated with lower horse-power and to prevent that only a power storage situation of
(Continued)



one of the plurality of power storage devices significantly degrades together with operation of the electrically-driven hydraulic work machine and besides, to extend a time period within which each of actuators of the electrically-driven hydraulic work machine can obtain a predetermined speed. Accordingly, a controller **50** includes a virtual limitation torque calculation section **51** and electric motor rotational speed control sections **52** and **53**. Variable horsepower control tables **52***r* and **53***r* are provided in the electric motor rotational speed control sections **52** and **53**, and limit values q**1***limit and q**2***limit for a virtual displacement of the variable horsepower control tables **52***r* and **53***r* are changed such that a charge state of a power storage device **170** and a charge state of another power storage device **270** become equal to each other.

8 Claims, 18 Drawing Sheets

(51) **Int. Cl.**

E02F 9/20	(2006.01)
F15B 11/17	(2006.01)
F15B 11/00	(2006.01)
E02F 3/32	(2006.01)

(52) **U.S. Cl.**

CPC ***E02F 9/2235*** (2013.01); ***E02F 9/2267*** (2013.01); ***E02F 9/2271*** (2013.01); ***E02F 9/2285*** (2013.01); ***E02F 9/2292*** (2013.01); ***E02F 9/2296*** (2013.01); ***F15B 11/00*** (2013.01); ***F15B 11/161*** (2013.01); ***F15B 11/17*** (2013.01); *E02F 3/325* (2013.01); *F15B 2211/20515* (2013.01); *F15B 2211/20538* (2013.01); *F15B 2211/20546* (2013.01); *F15B 2211/20576* (2013.01); *F15B 2211/253* (2013.01); *F15B 2211/255* (2013.01); *F15B 2211/275* (2013.01); *F15B 2211/575* (2013.01); *F15B 2211/605* (2013.01); *F15B 2211/6309* (2013.01); *F15B 2211/6313* (2013.01); *F15B 2211/6346* (2013.01); *F15B 2211/6651* (2013.01); *F15B 2211/7053* (2013.01); *F15B 2211/7058* (2013.01); *F15B 2211/71* (2013.01); *F15B 2211/7135* (2013.01)

(56) **References Cited**

U.S. PATENT DOCUMENTS

2005/0001567	A1	1/2005	Yoshimatsu	
2005/0001593	A1*	1/2005	Kawasumi	H02J 7/0071 320/132
2005/0060993	A1	3/2005	Oka et al.	
2010/0096918	A1	4/2010	Sawada et al.	
2014/0117934	A1*	5/2014	Kurikuma	B60L 58/13 320/109
2014/0227104	A1	8/2014	Takahashi et al.	

FOREIGN PATENT DOCUMENTS

JP	2008-109840	A	5/2008
JP	2008-256037	A	10/2008
JP	2015-198495	A	11/2015
WO	WO 2013/058326	A1	4/2013

OTHER PUBLICATIONS

International Search Report (PCT/ISA/210) issued in PCT Application No. PCT/JP2018/009830 dated May 22, 2018 with English translation (five pages).

Japanese-language Written Opinion (PCT/ISA/237) issued in PCT Application No. PCT/JP2018/009830 dated May 22, 2018 (three pages).

Korean-language Office Action issued in Korean Application No. 10-2019-7003923 dated Mar. 10, 2020 (six (6) pages).

* cited by examiner

FIG. 1

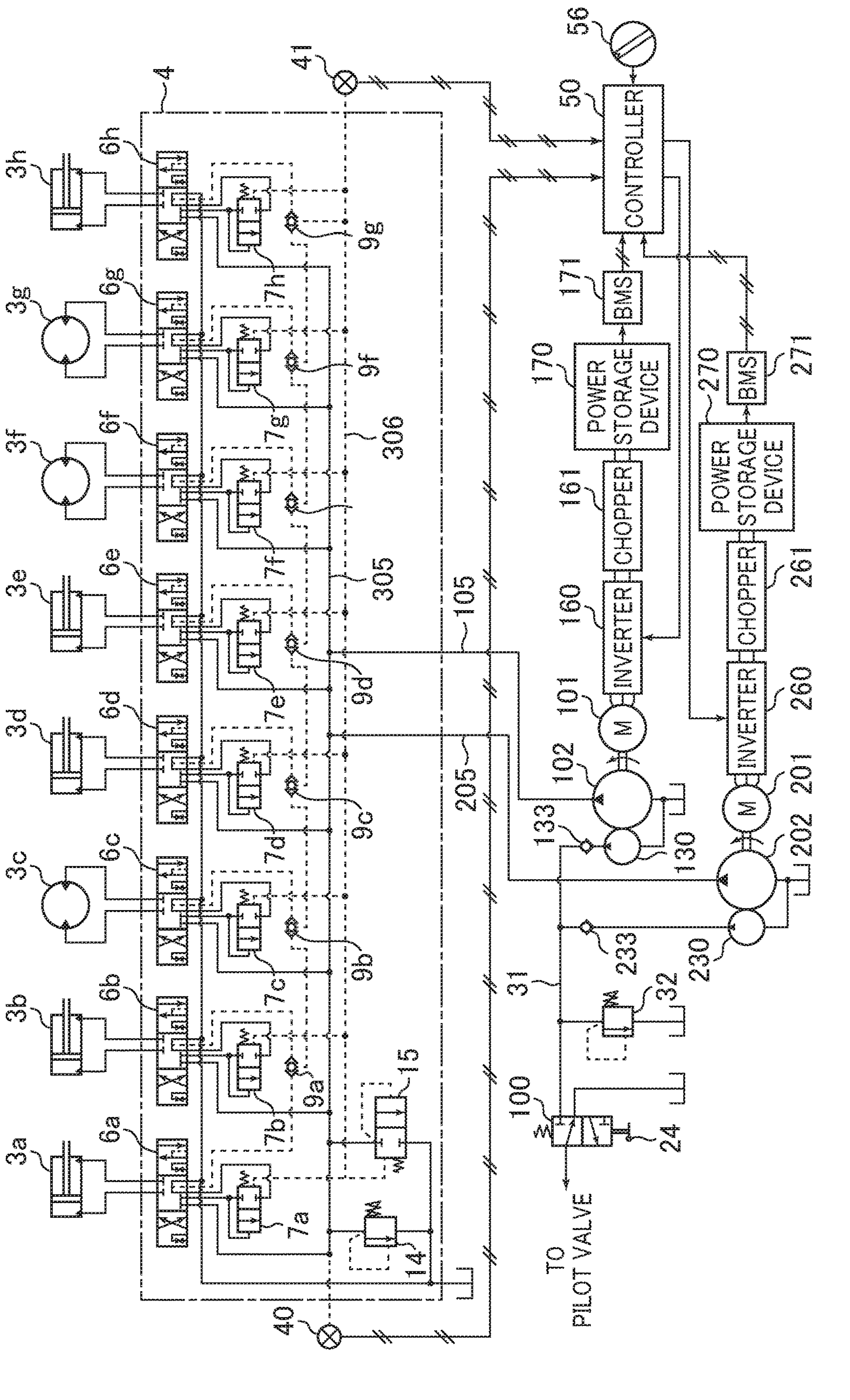

3a
3b
3c
3d
3e
3f
3g
3h
6a
6b
6c
6d
6e
6f
6g
6h
7a
7b
7c
7d
7e
7f
7g
7h
9a
9b
9c
9d
9f
9g
4
14
15
24
31
32
40
41
50
CONTROLLER
56
100
101
M
102
105
130
133
160
INVERTER
161
CHOPPER
170
POWER STORAGE DEVICE
171
BMS
201
M
202
205
230
233
260
INVERTER
261
CHOPPER
270
POWER STORAGE DEVICE
271
BMS
305
306
TO PILOT VALVE

FIG.2

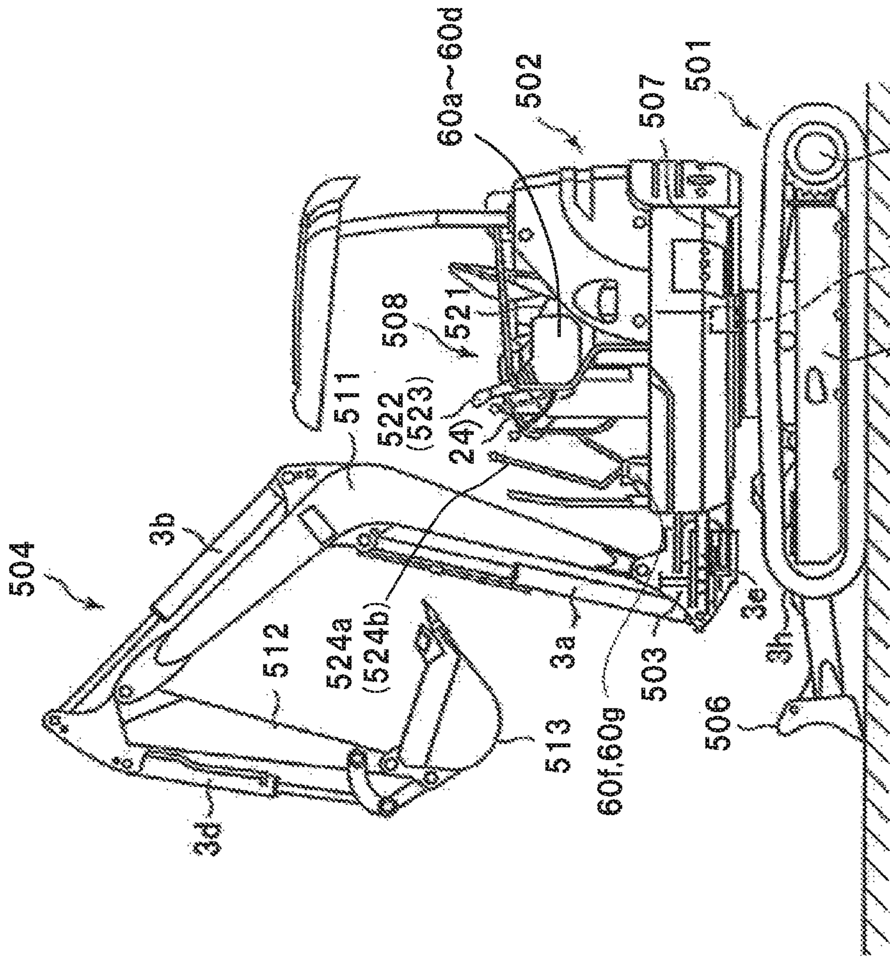

504
3b
511
508
521
522
(523)
24
60a~60d
502
507
501
3f (3g)
3c
505
512
524a
(524b)
3a
3e
3h
503
60f,60g
506
513
3d

FIG. 3

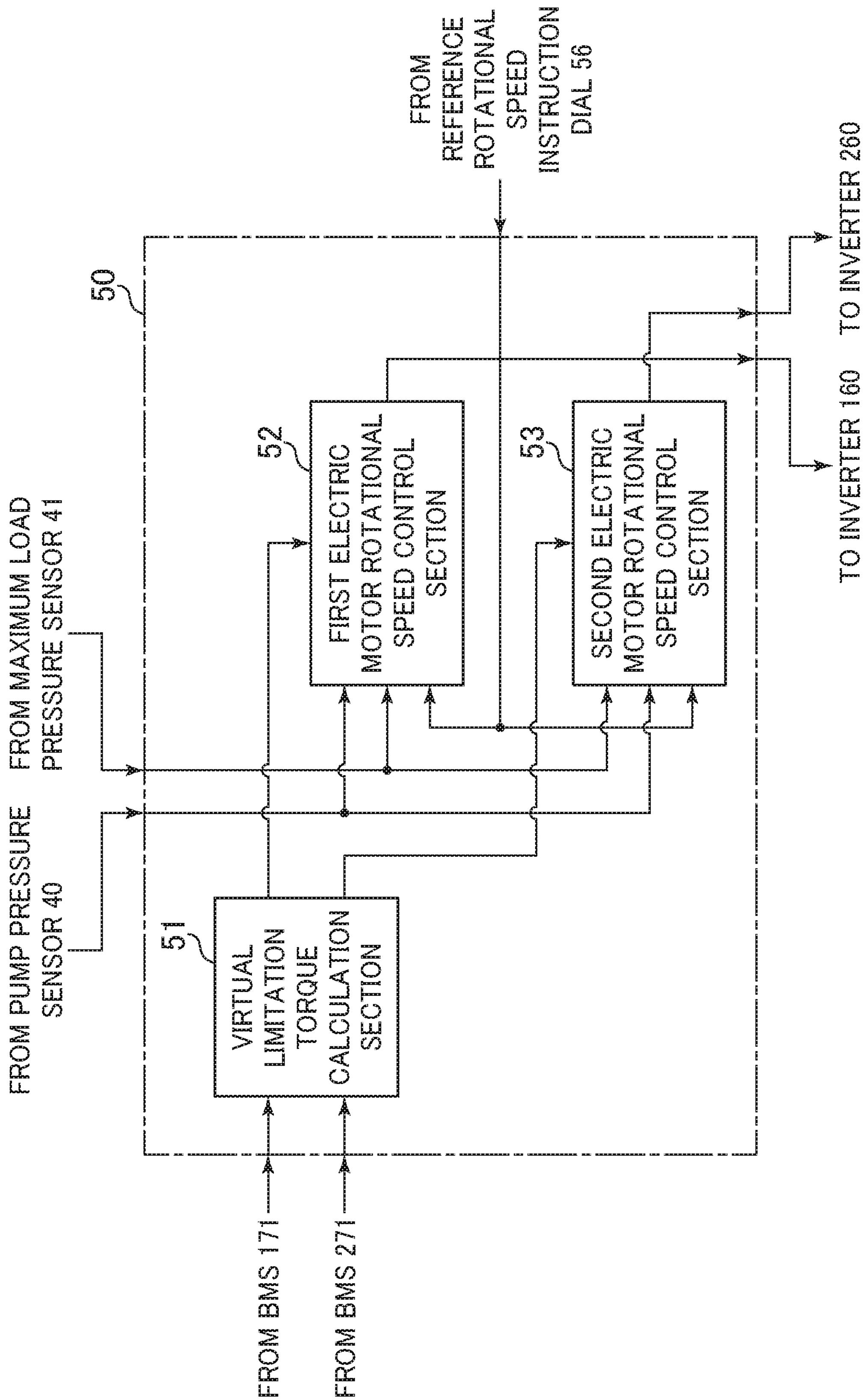
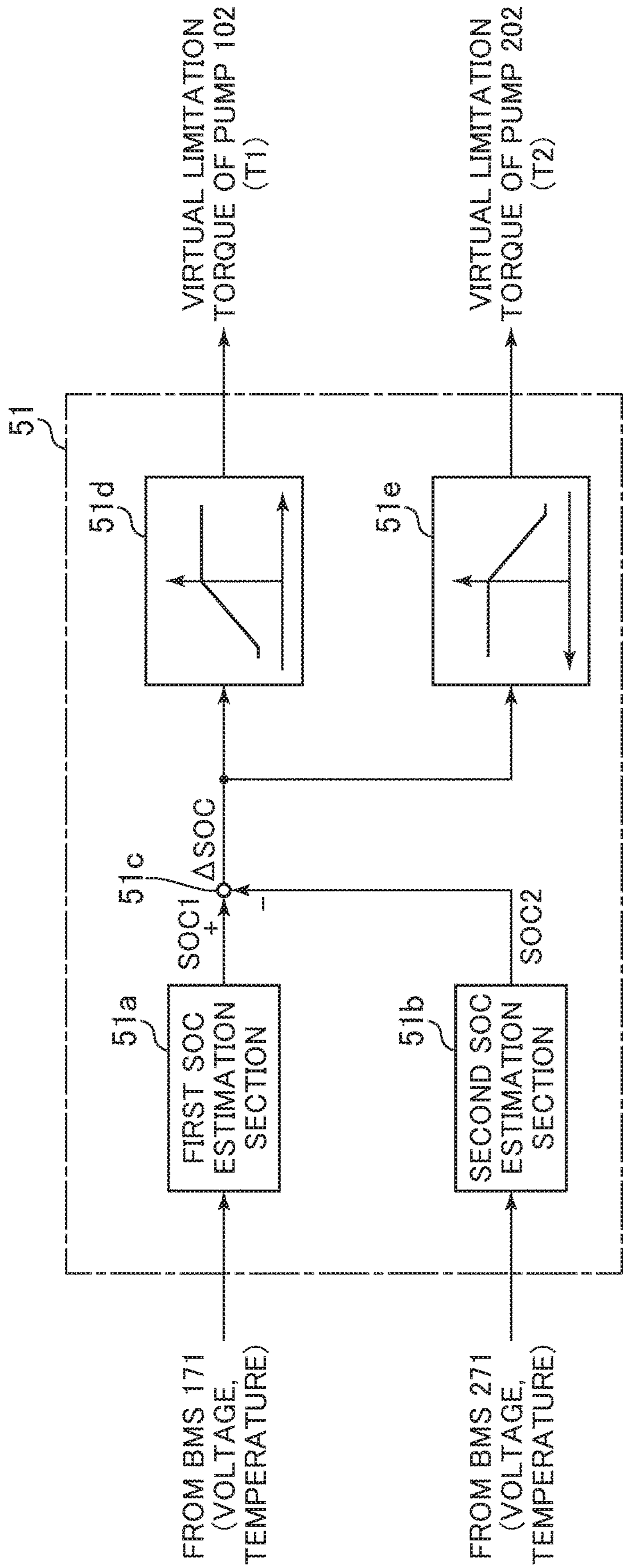

50
51
VIRTUAL LIMITATION TORQUE CALCULATION SECTION
52
FIRST ELECTRIC MOTOR ROTATIONAL SPEED CONTROL SECTION
53
SECOND ELECTRIC MOTOR ROTATIONAL SPEED CONTROL SECTION
FROM PUMP PRESSURE SENSOR 40
FROM MAXIMUM LOAD PRESSURE SENSOR 41
FROM REFERENCE ROTATIONAL SPEED INSTRUCTION DIAL 56
TO INVERTER 260
TO INVERTER 160
FROM BMS 171
FROM BMS 271

FIG. 4



51
FROM BMS 171 (VOLTAGE, TEMPERATURE)
FROM BMS 271 (VOLTAGE, TEMPERATURE)
51a
FIRST SOC ESTIMATION SECTION
51b
SECOND SOC ESTIMATION SECTION
SOC1
SOC2
+
−
51c
ΔSOC
51d
51e
VIRTUAL LIMITATION TORQUE OF PUMP 102 (T1)
VIRTUAL LIMITATION TORQUE OF PUMP 202 (T2)

FIG.5A

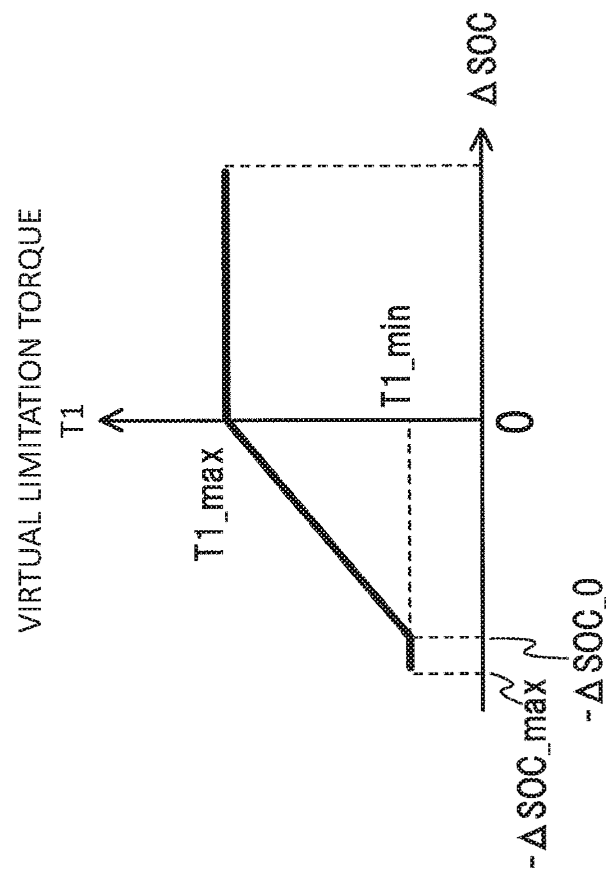

VIRTUAL LIMITATION TORQUE
T1
T1_max
T1_min
0
Δ SOC
- Δ SOC_0
- Δ SOC_max

FIG.5B

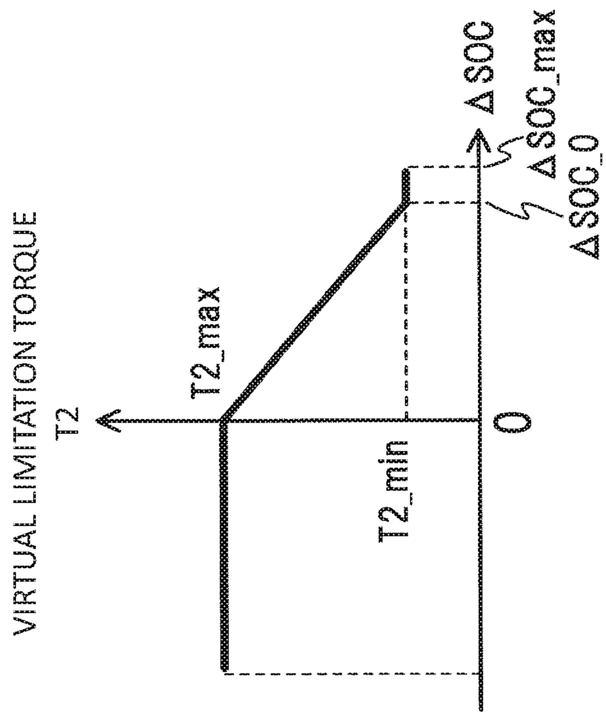

VIRTUAL LIMITATION TORQUE
T2
T2_max
T2_min
0
Δ SOC
Δ SOC_max
Δ SOC_0

FIG. 6

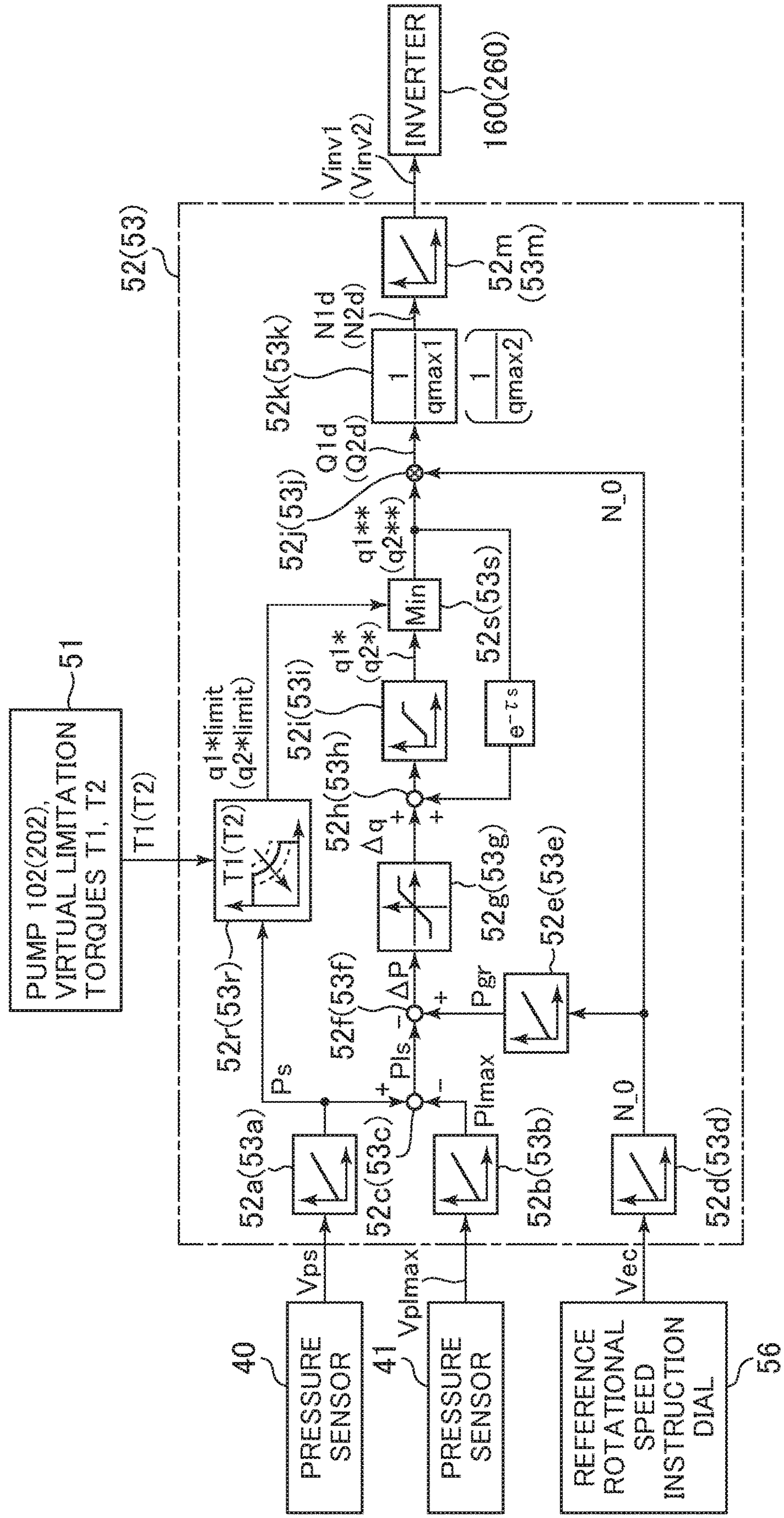

PUMP 102(202), VIRTUAL LIMITATION TORQUES T1, T2
51
T1(T2)
52(53)
INVERTER
160(260)
Vinv1
(Vinv2)
52m
(53m)
N1d
(N2d)
52k(53k)
1
qmax1
1
qmax2
Q1d
(Q2d)
52j(53j)
q1**
(q2**)
N_0
52s(53s)
Min
q1*limit
(q2*limit)
q1*
(q2*)
52i(53i)
e−τs
52h(53h)
Δq
52g(53g)
52e(53e)
52r(53r)
52f(53f)
ΔP
Pgr
Ps
Pls
Plmax
52a(53a)
52c(53c)
52b(53b)
52d(53d)
Vps
Vplmax
Vec
40
PRESSURE SENSOR
41
PRESSURE SENSOR
REFERENCE ROTATIONAL SPEED INSTRUCTION DIAL
56

FIG. 7

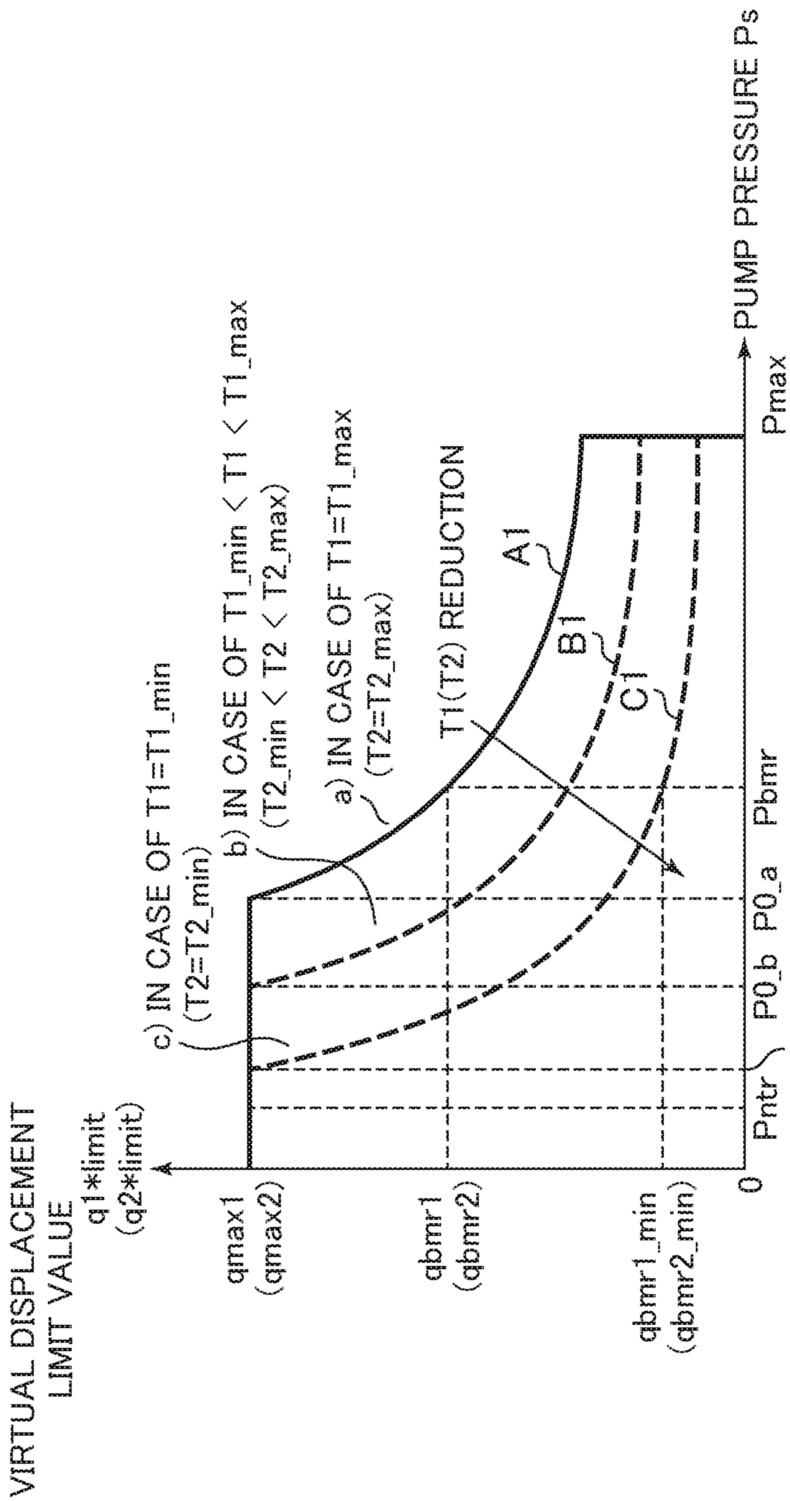

VIRTUAL DISPLACEMENT LIMIT VALUE
q1*limit
(q2*limit)
qmax1
(qmax2)
qbmr1
(qbmr2)
qbmr1_min
(qbmr2_min)
0
Pntr
P0_c
P0_b
P0_a
Pbmr
Pmax
PUMP PRESSURE Ps
c) IN CASE OF T1=T1_min
(T2=T2_min)
b) IN CASE OF T1_min < T1 < T1_max
(T2_min < T2 < T2_max)
a) IN CASE OF T1=T1_max
(T2=T2_max)
T1(T2) REDUCTION
A1
B1
C1

FIG. 8

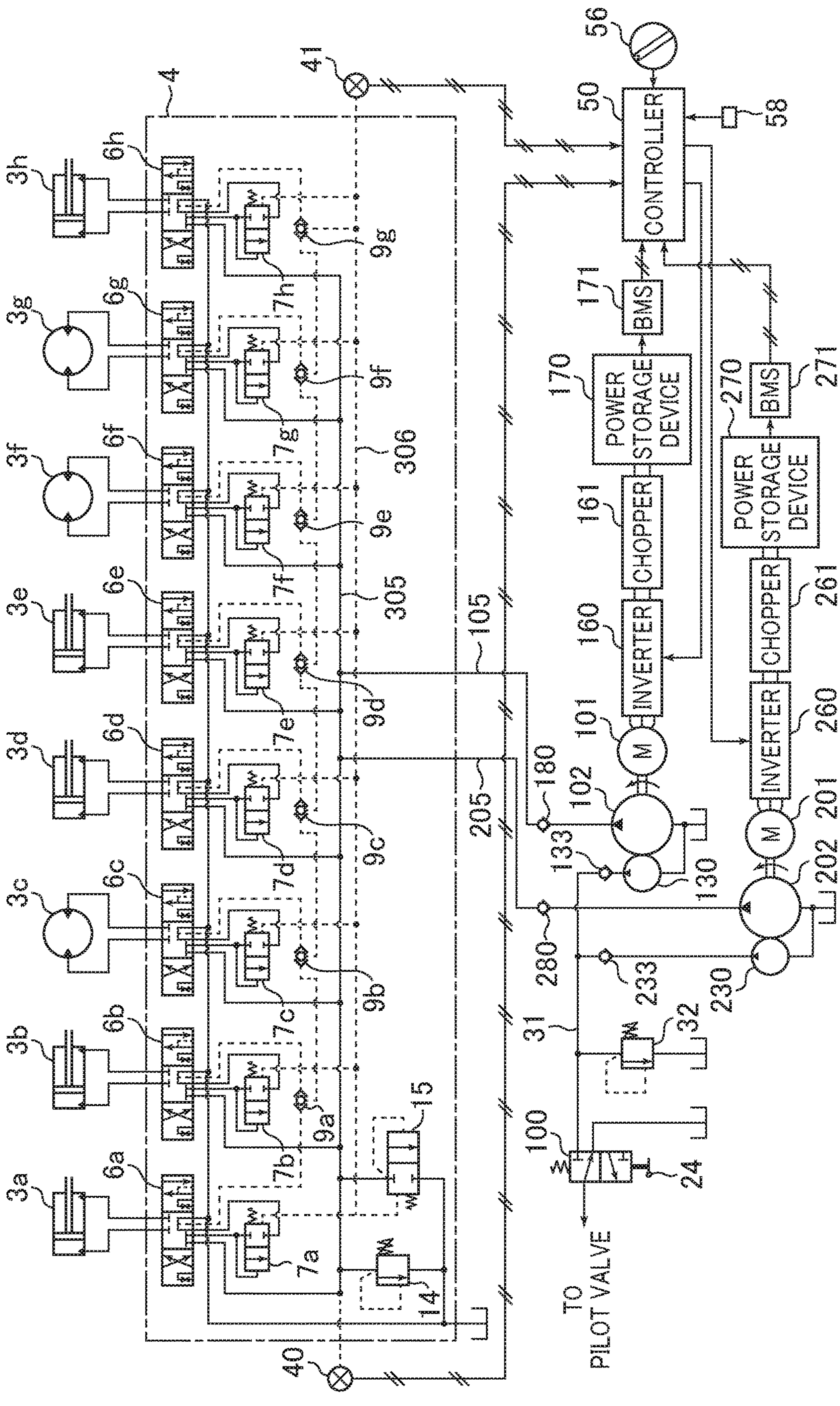

3a
3b
3c
3d
3e
3f
3g
3h
6a
6b
6c
6d
6e
6f
6g
6h
7a
7b
7c
7d
7e
7f
7g
7h
9a
9b
9c
9d
9e
9f
9g
4
14
15
24
31
32
40
41
50
56
58
100
101
102
105
130
133
160
161
170
171
180
201
202
205
230
233
260
261
270
271
280
305
306
CONTROLLER
BMS
POWER STORAGE DEVICE
CHOPPER
INVERTER
M
TO PILOT VALVE

FIG. 9

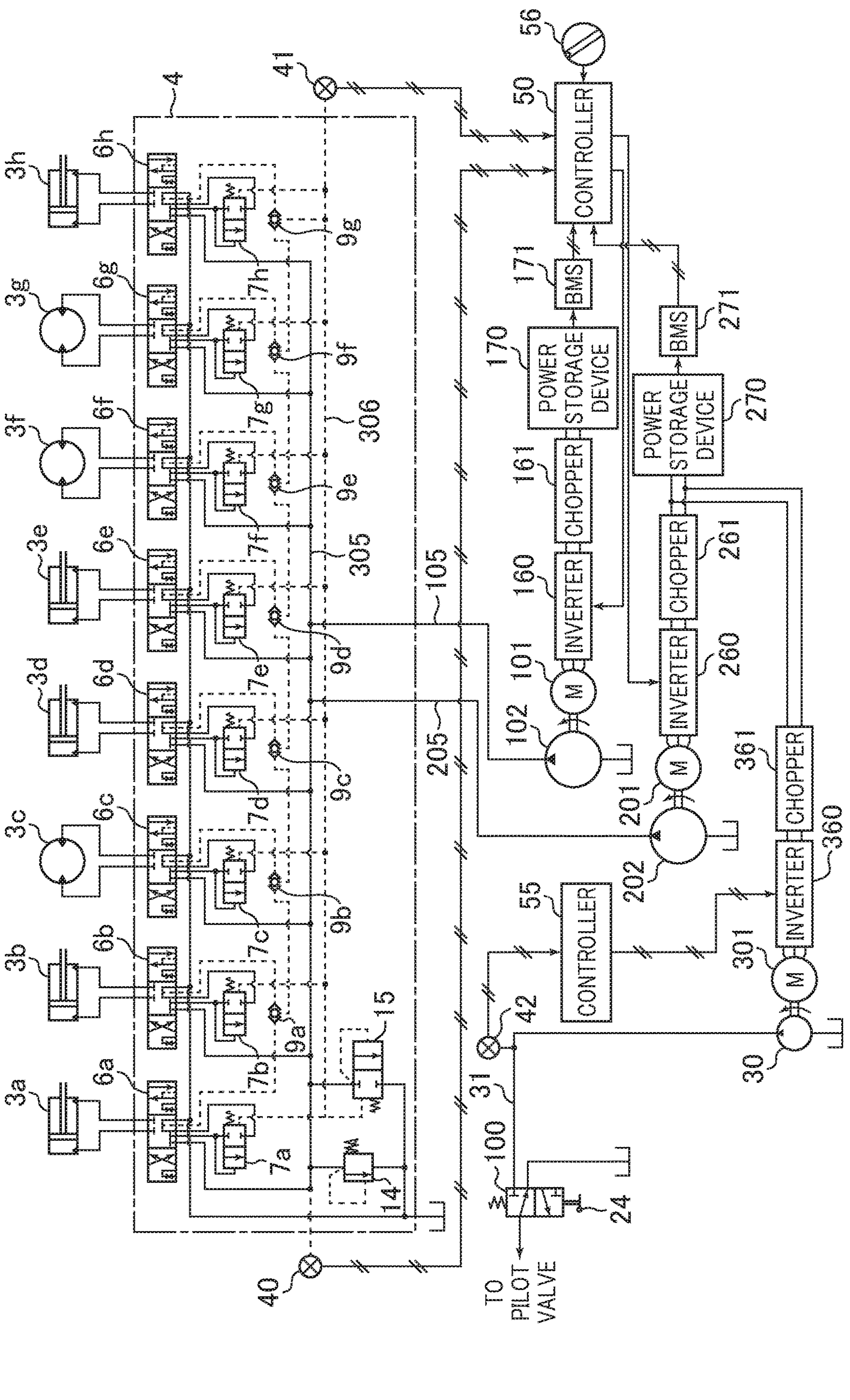

CONTROLLER
POWER STORAGE DEVICE
BMS
CHOPPER
INVERTER
M
CONTROLLER
INVERTER
CHOPPER
TO PILOT VALVE

FIG. 10

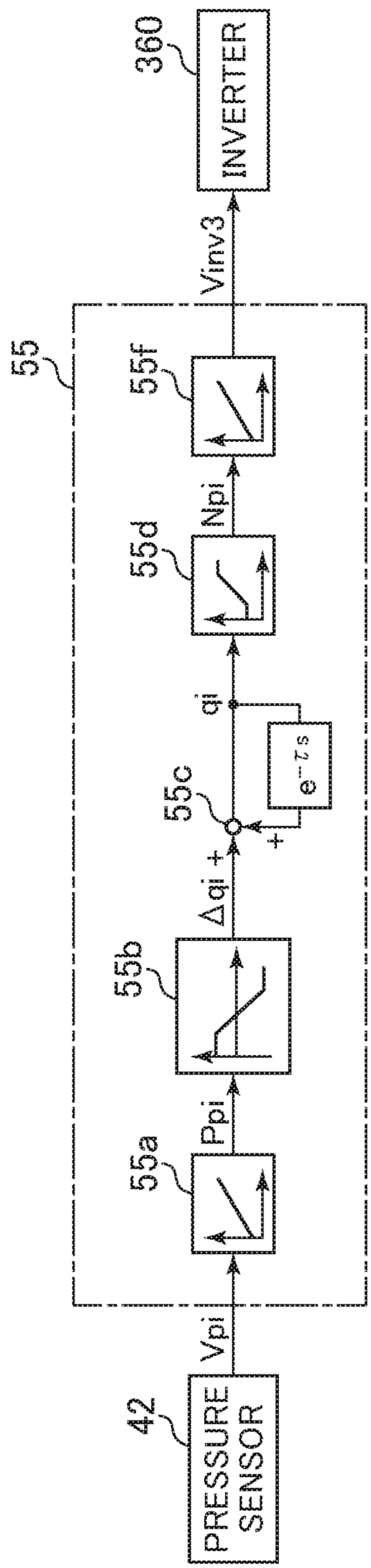

360
INVERTER
Vinv3
55
55f
Npi
55d
qi
$e^{-\tau s}$
55c
+
Δqi +
55b
Ppi
55a
Vpi
42
PRESSURE SENSOR

FIG.11

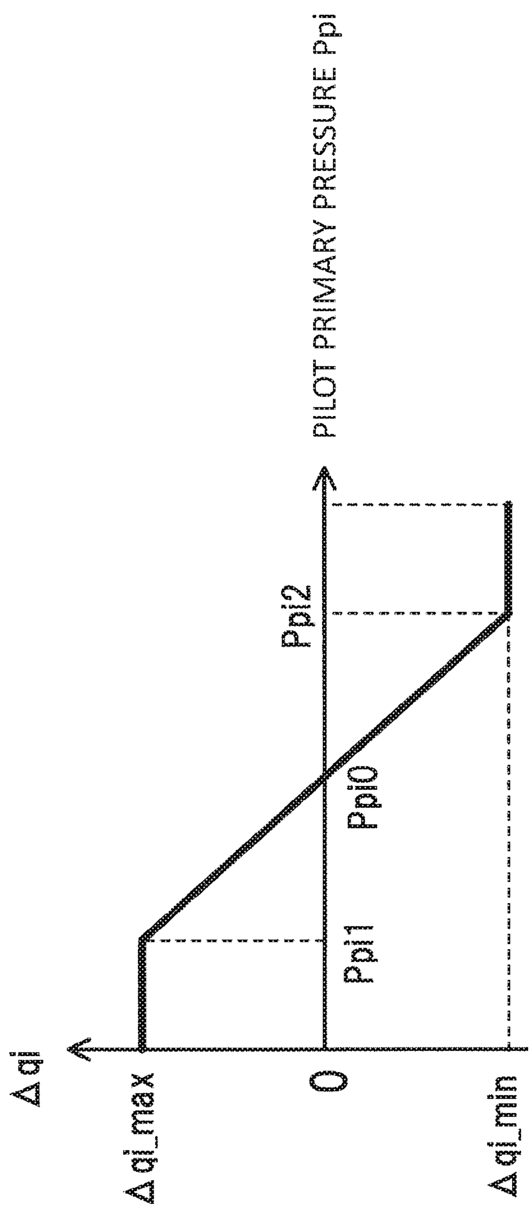

PILOT PRIMARY PRESSURE Ppi
Ppi2
Ppi0
Ppi1
Δqi
Δqi_max
0
Δqi_min

FIG.12B

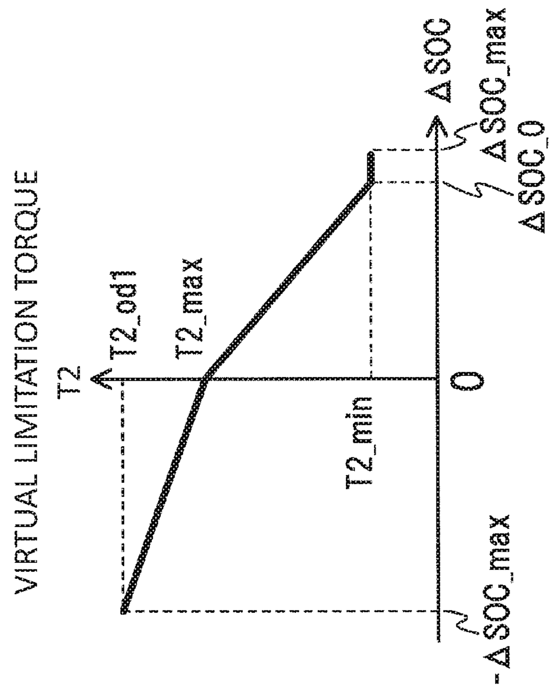


FIG.12A

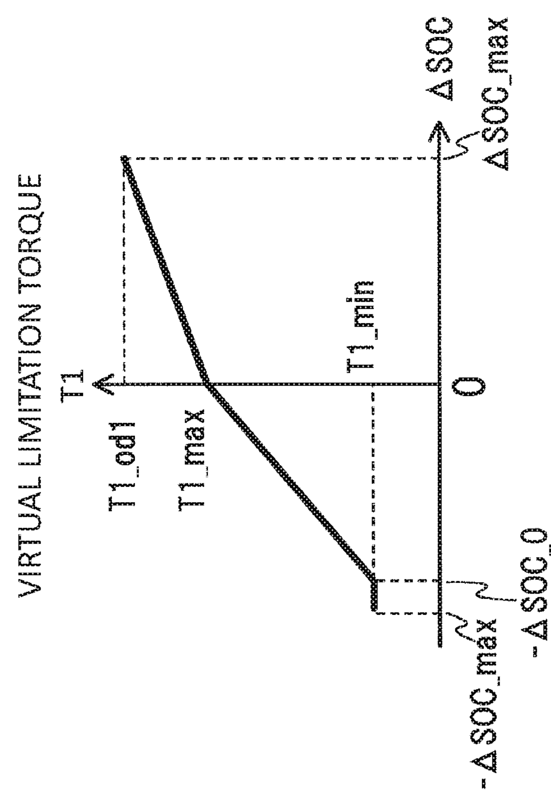


FIG.12D

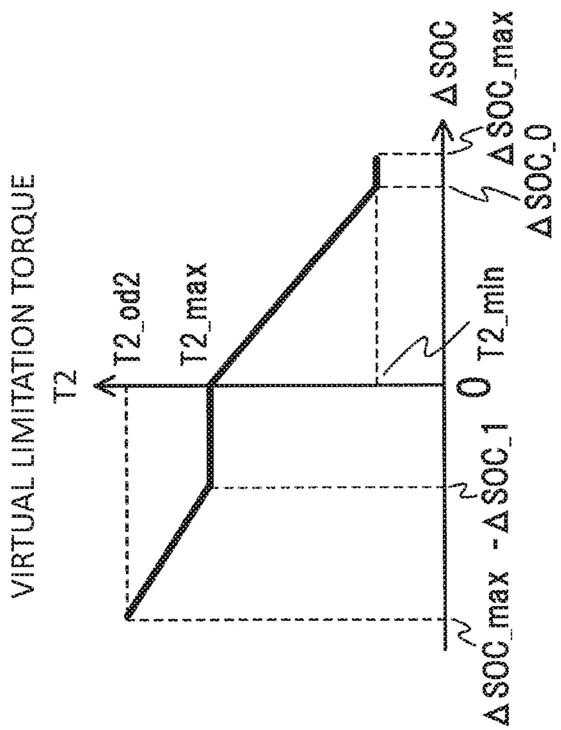


FIG.12C

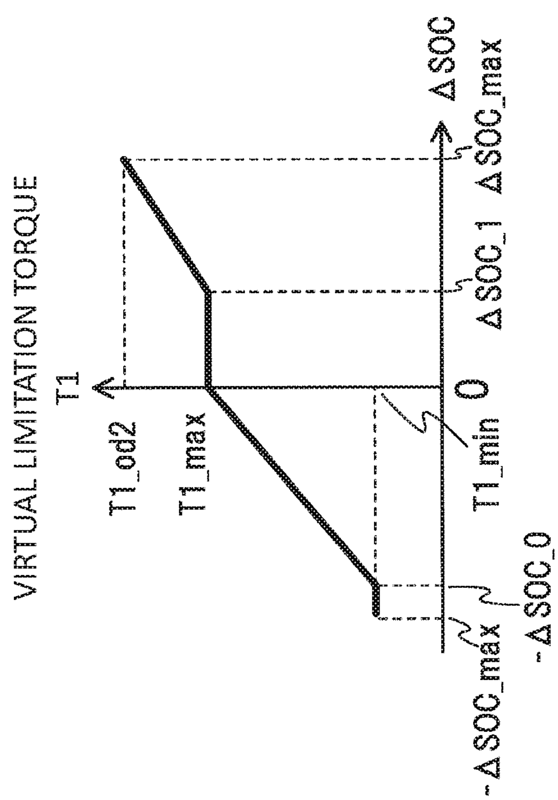


FIG. 13

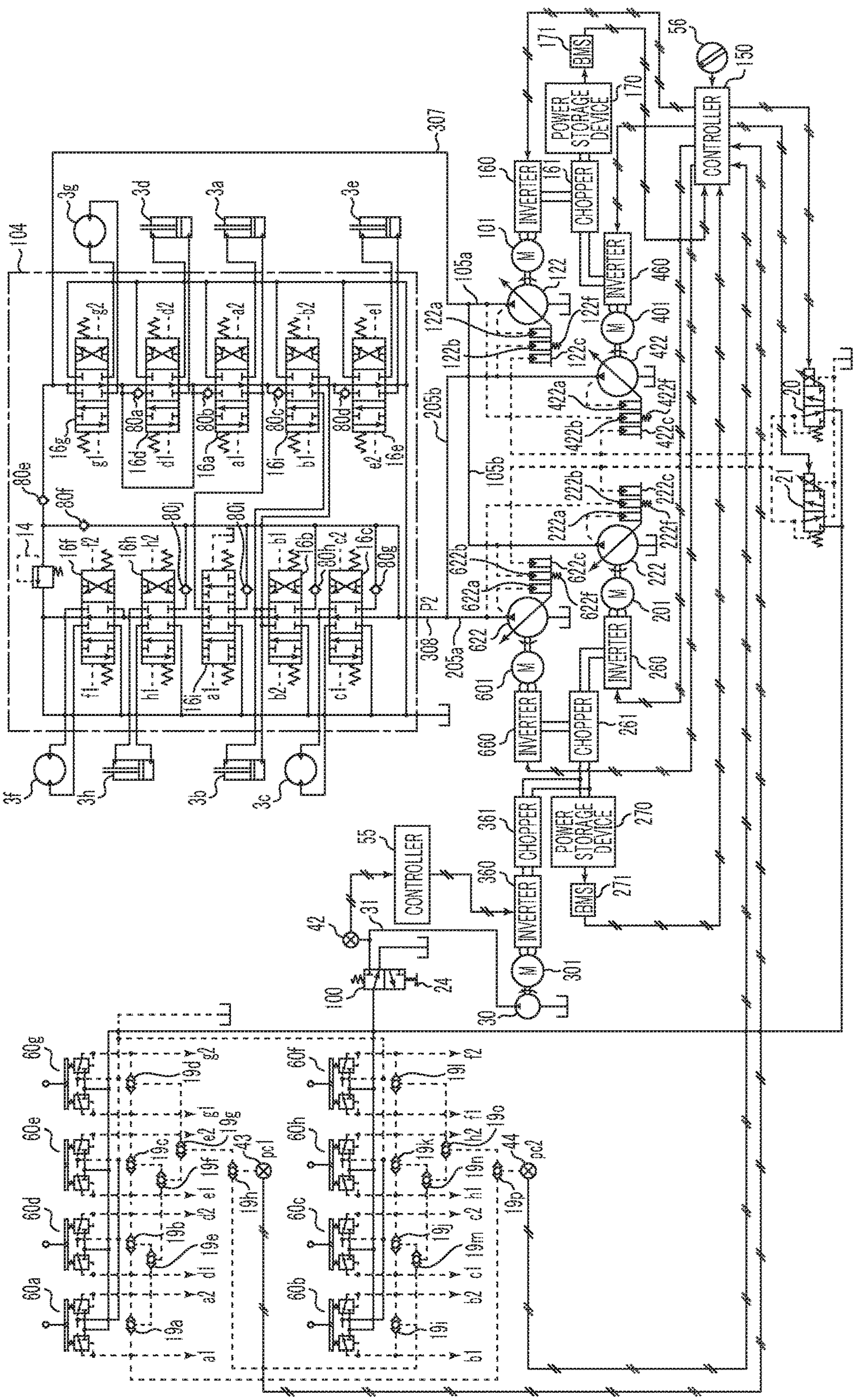

CONTROLLER
INVERTER
CHOPPER
POWER STORAGE DEVICE
BMS
CONTROLLER
INVERTER
CHOPPER
POWER STORAGE DEVICE
BMS
INVERTER
INVERTER
CHOPPER
INVERTER

FIG. 14

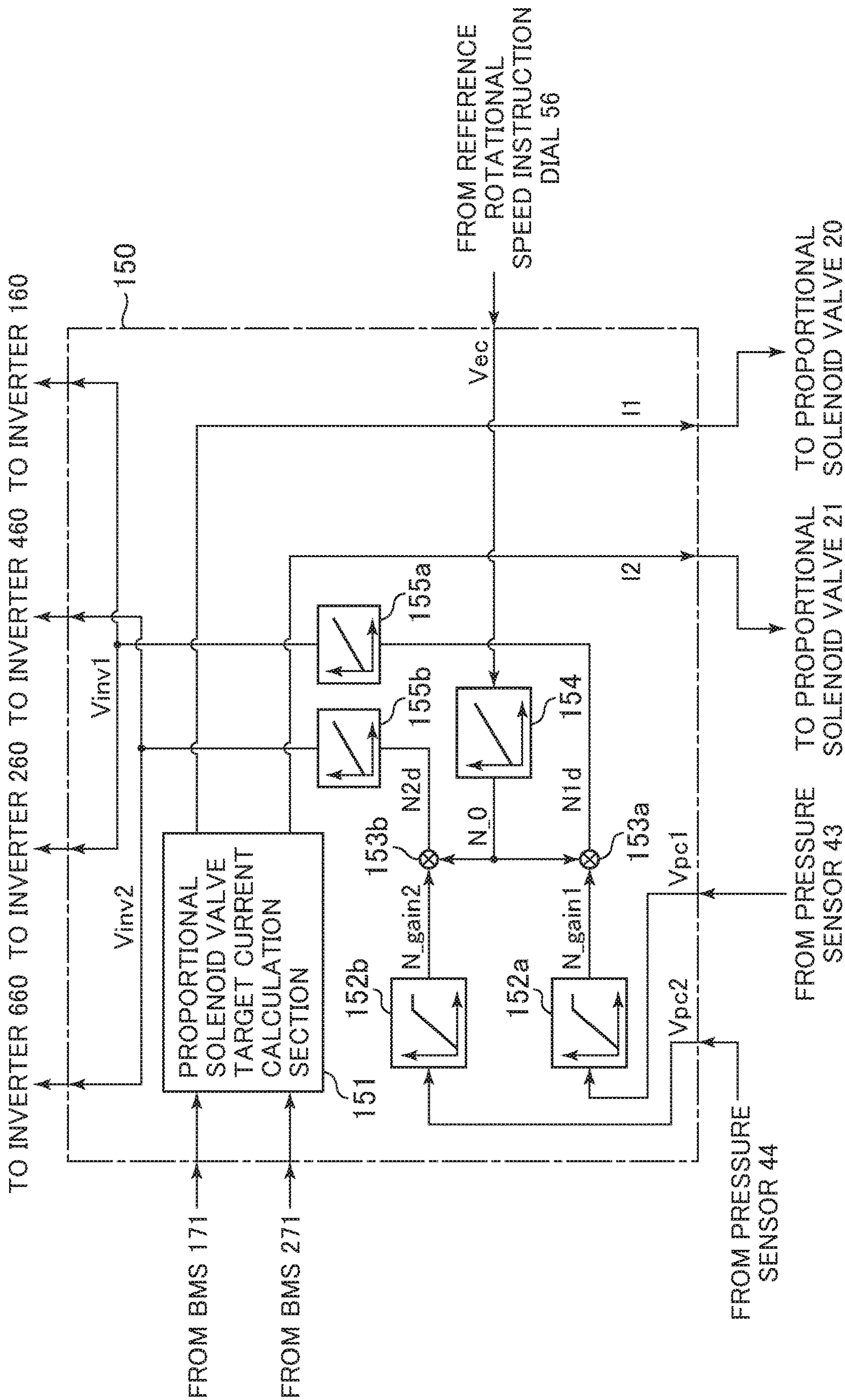

TO INVERTER 660
TO INVERTER 260
TO INVERTER 460
TO INVERTER 160
150
FROM REFERENCE ROTATIONAL SPEED INSTRUCTION DIAL 56
Vec
I1
I2
TO PROPORTIONAL SOLENOID VALVE 20
TO PROPORTIONAL SOLENOID VALVE 21
Vinv1
Vinv2
155a
155b
154
PROPORTIONAL SOLENOID VALVE TARGET CURRENT CALCULATION SECTION
151
N2d
N1d
N_0
153b
153a
N_gain2
N_gain1
152b
152a
Vpc1
Vpc2
FROM PRESSURE SENSOR 43
FROM PRESSURE SENSOR 44
FROM BMS 171
FROM BMS 271

FIG. 15

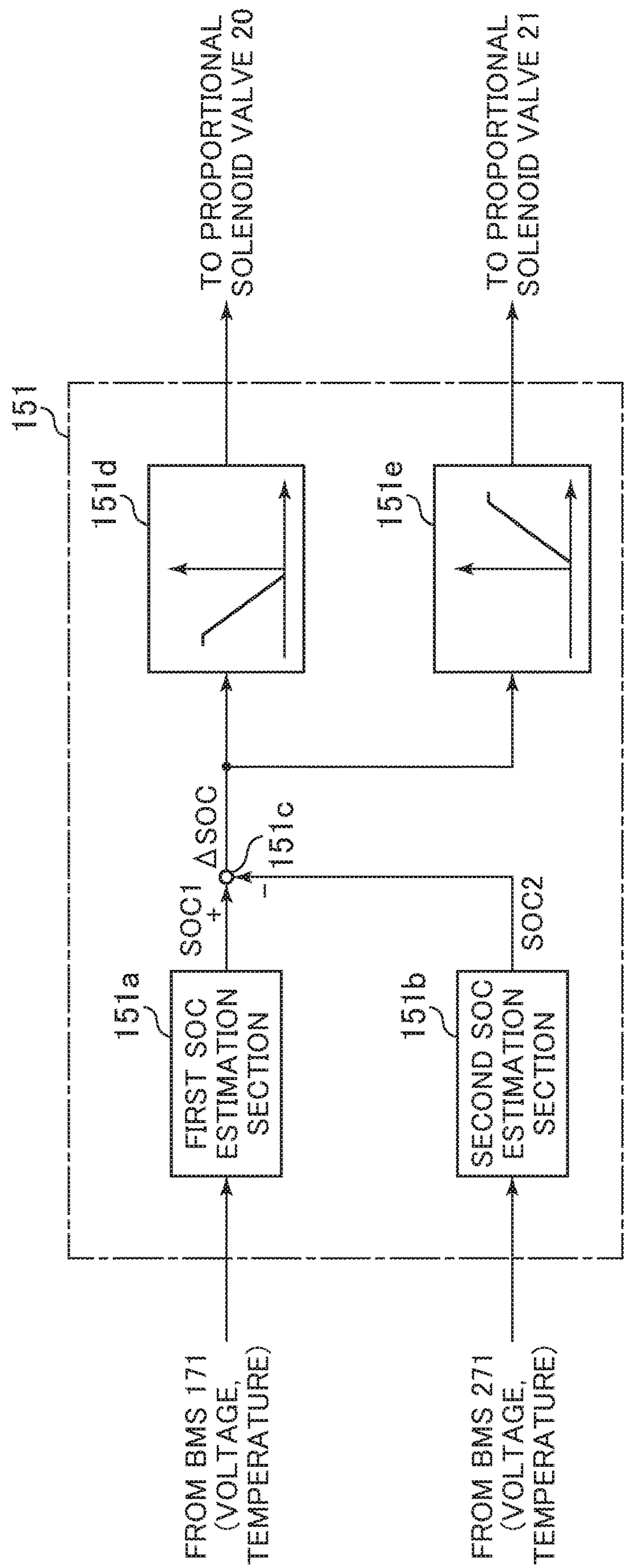

TO PROPORTIONAL SOLENOID VALVE 20
TO PROPORTIONAL SOLENOID VALVE 21
151
151d
151e
ΔSOC
151c
SOC1
+
−
SOC2
151a
FIRST SOC ESTIMATION SECTION
151b
SECOND SOC ESTIMATION SECTION
FROM BMS 171 (VOLTAGE, TEMPERATURE)
FROM BMS 271 (VOLTAGE, TEMPERATURE)

FIG.16B

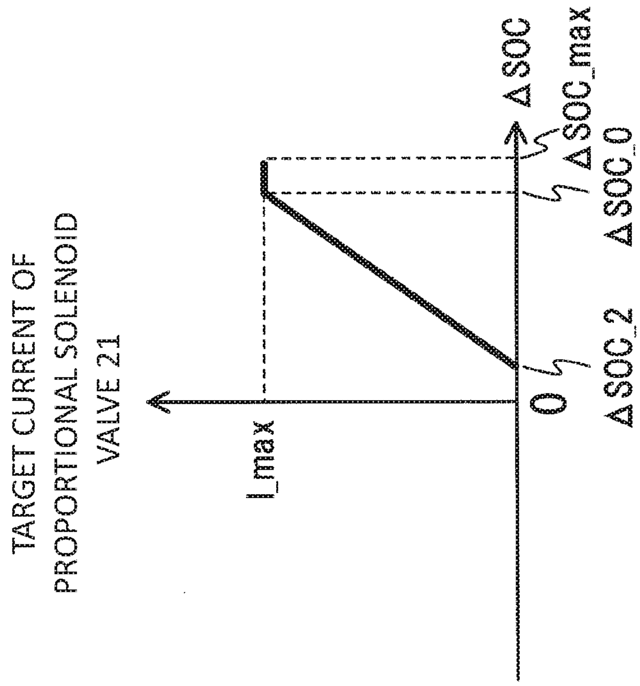


FIG.16A

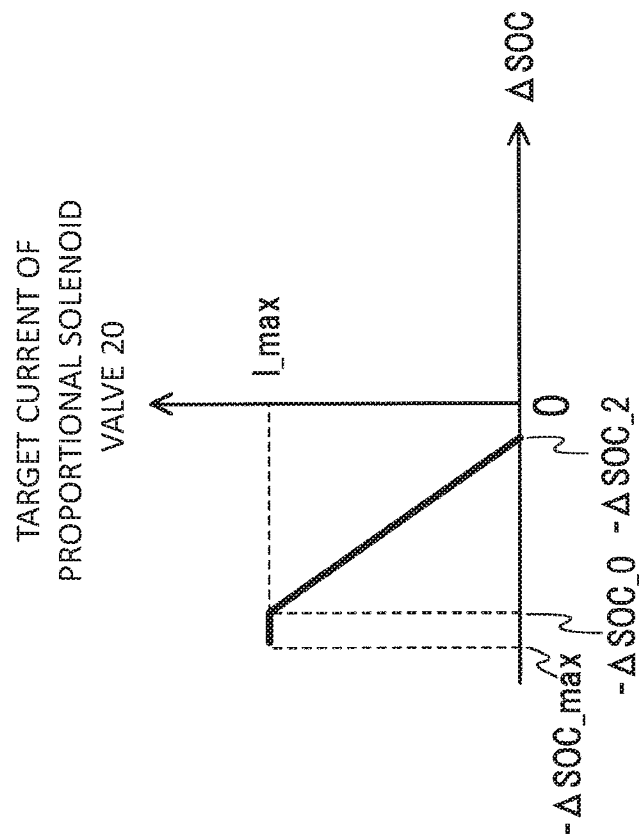
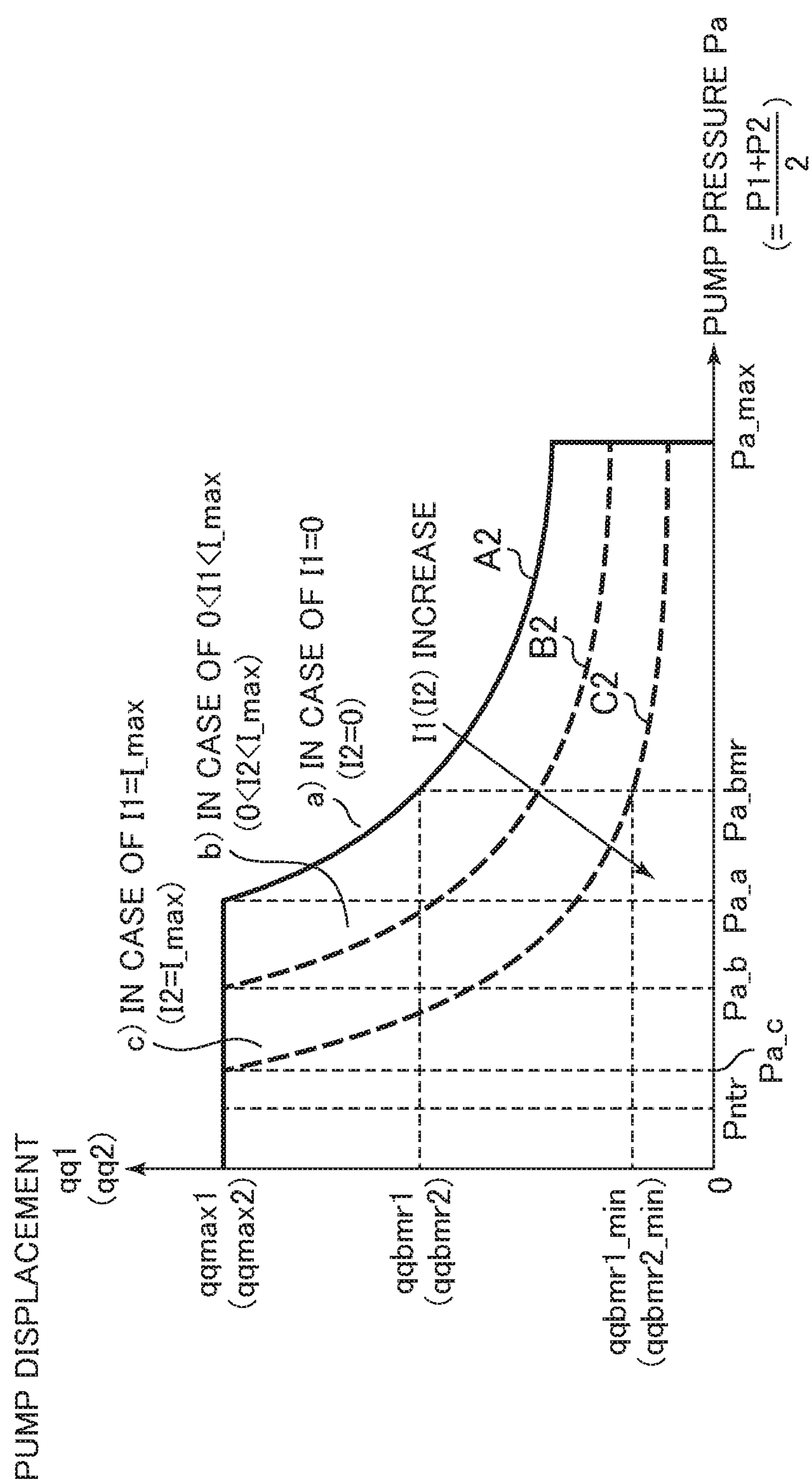


FIG. 17


PUMP DISPLACEMENT
qq1
(qq2)
qqmax1
(qqmax2)
qqbmr1
(qqbmr2)
qqbmr1_min
(qqbmr2_min)
0
Pntr
Pa_c
Pa_b
Pa_a
Pa_bmr
Pa_max
PUMP PRESSURE Pa
(= (P1+P2)/2)
c) IN CASE OF I1=I_max
(I2=I_max)
b) IN CASE OF 0<I1<I_max
(0<I2<I_max)
a) IN CASE OF I1=0
(I2=0)
I1(I2) INCREASE
A2
B2
C2

HYDRAULIC DRIVE SYSTEM FOR ELECTRICALLY-DRIVEN HYDRAULIC WORK MACHINE

TECHNICAL FIELD

The present invention relates to a hydraulic drive system for an electrically-driven hydraulic work machine such as a hydraulic excavator in which a hydraulic pump is driven by an electric motor to drive actuators to perform various works, and particularly to the hydraulic drive system that controls a rotational speed of an electric motor for driving the hydraulic pump such that absorption torque of the hydraulic pump becomes equal to or lower than a certain value, thereby to perform so-called horsepower control.

BACKGROUND ART

A conventional technology of an electrically-driven hydraulic work machine such as a hydraulic excavator in which a hydraulic pump is driven by an electric motor to drive actuators to perform various works is disclosed in Patent Document 1 and Patent Document 2.

According to Patent Document 1, a configuration is proposed in which a hydraulic pump of fixed displacement type driven by an electric motor is provided and the rotational speed of the electric motor is controlled such that a differential pressure between a delivery pressure of the hydraulic pump and a maximum load pressure of a plurality of actuators is fixed to perform load sensing control. If this technology is used, then the number of hydraulic pipes can be decreased from that, in an alternative case in which the load sensing control is performed using a hydraulic pump of variable displacement type, and it is facilitated to apply high-efficiency load sensing control to a small-sized hydraulic excavator or the like in which the necessary installation space is small.

Meanwhile, according to Patent Document 2, an electrically-driven hydraulic work machine is proposed in which a hydraulic pump is configured as that of the variable displacement type including only a horsepower controlling function and a controlling algorism that simulates a horsepower control characteristic of a hydraulic pump of the variable displacement type is provided in a controller for controlling the rotational speed of an electric motor. If this technology is used, then a power storage device that is a power source for the electric motor can be made long-lasting and besides, the electric motor can be downsized in addition to the advantages of Patent Document 1.

PRIOR ART DOCUMENT

Patent Document

Patent Document 1: JP-2008-256037-A

Patent Document 2: WO 2013/058326

SUMMARY OF THE INVENTION

Problem to be Solved by the Invention

However, also in the case where the conventional technologies disclosed in Patent Document 1 and Patent Document 2 are used, there is such a problem as described below.

In general, if horsepower required for an electrically-driven hydraulic work machine increases, also an electric motor having high rated output power must be selected in response to the increase of the horsepower. However, if the

rated output power of the electric motor increases, then there is a case in which the rated voltage becomes high in order to avoid current that flows in the electric motor and an inverter, which is a controlling circuit for the electric motor, from becoming high (for example, while the rated voltage where the rated output power is 20 kW is 200 V, the rated voltage where the rated output power is 40 kW is 400 V).

In this case, the rated voltage of a power storage device or the rated voltage of a step-up/step-down chopper cannot be made common to that of the electrically-driven hydraulic work machine that is capable of being operated with lower horsepower and it is necessary to set rated voltages of a power storage device and a step-up/step-down chopper for exclusive use, and this is cumbersome.

As a countermeasure for avoidance of the cumbersomeness, it is conceivable to provide a plurality of systems of a hydraulic pump, an electric motor, an inverter, a step-up/step-down chopper, a power storage device, and so forth in parallel to one another such that hydraulic fluids delivered from the plurality of systems of the pumps are merged and supplied to a plurality of flow control valves for controlling hydraulic supplied to fluids to the actuators.

According to the countermeasure, since hydraulic fluids delivered from the hydraulic pumps of the plurality of systems are merged, the actuators can be operated at a predetermined speed, and the rated voltage of the power storage device or the rated voltage of the step-up/step-down chopper can be made common to that of the electrically-driven hydraulic work machine that is capable of being operated with lower horsepower.

However, also in the case where such a configuration as described above is adopted, there is a problem in the following cases.

First, since a plurality of systems of a hydraulic pump, an electric motor, an inverter, a step-up/step-down chopper, and a power storage device are provided in parallel to one another, even in the case where the plurality of systems of the hydraulic pumps have same specifications and generate equal power, a small difference sometimes occurs in electric power consumption of the power storage devices due to the machine efficiency of the plurality of systems of the hydraulic pumps, efficiencies of the inverters and step-up/step-down choppers, and so forth.

Second, although the power storage device is in most cases configured normally by connecting a plurality of cells in series, voltages of the cells have some dispersion, and the voltage of a cell whose voltage decreases most from among the plurality of cells has a strong influence on the charge state (=SOC (State of Charge)) of the power storage device. Therefore, in the case where a plurality of systems of power storage devices are provided as described above, even if the plurality of systems of power storage devices should consume fully equal electric power, a different SOC is sometimes indicated due to a difference in dispersion of the voltages of the plurality of cells configuring each of the plurality of systems of power storage devices.

In such first and second cases as described above, when the electrically-driven hydraulic work machine is continuously operated, unbalance occurs in the SOC of the plurality of systems of power storage devices described above and, as a result, the SOC of one of the power storage devices becomes lower than a minimum usable level earlier, which disables the power storage device, in some cases.

In such a case as just described, there is a problem that one of the plurality of systems of power storage devices is disabled and the hydraulic pump that has been driven by an electric motor to which electric power has been supplied by the disabled power storage device stops the supply of hydraulic fluid, which significantly decreases the actuator speed of each of the actuators of the hydraulic work machine and hence the workability of the hydraulic work machine.

It is an object of the present invention to provide a hydraulic drive system for an electrically-driven hydraulic work machine in which a plurality of power storage devices, a plurality of electric motors, and a plurality of hydraulic pumps are provided in parallel to one another such that the rated voltage of each of various electric equipment such as power storage devices can be made common to that of an electrically-driven hydraulic work machine that is capable of being operated with lower horsepower and it is prevented that only the power storage situation of one of the plurality of power storage devices significantly degrades together with operation of the electrically-driven hydraulic work machine and besides, the time period within which each of the actuators of the electrically-driven hydraulic work machine can obtain a predetermined speed can be extended.

Means for Solving the Problem

In order to solve the problems described above, according to the present invention, a hydraulic drive system for an electrically-driven hydraulic work machine including a first hydraulic pump, a plurality of actuators driven by hydraulic fluid supplied from the first hydraulic pump, a plurality of flow control valves that control directions and flow rates of the hydraulic fluids to be supplied to the plurality of actuators, a first electric motor that drives the first hydraulic pump, a first power storage device for supplying electric power to the first electric motor, and a first horsepower control device configured to decrease, when a delivery pressure of the first hydraulic pump increases, a delivery flow rate of the first hydraulic pump to control an absorption horsepower of the first hydraulic pump so as not to exceed a first allowable value, the hydraulic drive system comprising a second hydraulic pump, a common hydraulic fluid supply line in which hydraulic fluids delivered from the first and second hydraulic pumps are merged and are supplied to the plurality of flow control valves, a second electric motor that drives the second hydraulic pump, a second power storage device that supplies electric power to the second electric motor, a second horsepower control device configured to decreases, when a delivery pressure of the second hydraulic pump increases, a delivery flow rate of the second hydraulic pump to control an absorption horsepower of the second hydraulic pump so as not to exceed a second allowable value, and a controller including a horsepower distribution control section configured to change at least one of the first and second allowable values of the first and second horsepower control devices such that a charge state of the first power storage device and a charge state of the second power storage device become equal to each other.

Since the second hydraulic pump, second electric motor, second power storage device, and the common hydraulic fluid supply line in which hydraulic fluids delivered from the first and second hydraulic pumps are merged are provided in addition to the first hydraulic pump, first electric motor, and first power storage device such that the merged hydraulic fluids are supplied to the plurality of flow control valves and then are further supplied to the plurality of actuators in such a manner as described above, the rated voltage of various electric equipment such as the power storage devices can be made common to that of an electrically-driven hydraulic work machine that is capable of being operated with lower horse power.

Further, by providing the controller including the horsepower distribution control section that changes at least one of the first and second allowable values of the first and second horsepower control devices such that the charge state of the first power storage device and the charge state of the second power storage device become equal to each other, even in the case where there is a difference between machine efficiencies of the first and second hydraulic pumps (plurality of hydraulic pumps) or in efficiency of electric equipment such as inverters or step-up/step-down choppers that control the rotational speed of the plurality of electric motors for individually driving the plurality of hydraulic pumps or even in the case where there is a difference in electric power consumption amount or charge state characteristic of the plurality of power storage devices, the difference gradually decrease while the charge states of the plurality of power storage devices are controlled so as to become equal to each other. Therefore, it is prevented that the power storage situation of only one of the plurality of power storage devices significantly degrades, and the time period within which each of the actuators of the electrically-driven hydraulic work machine obtain a predetermined speed can be extended.

Effect of the Invention

According to the present invention, the rated voltage of each of various electric equipment such as a power storage device can be made common to that of an electrically-driven hydraulic work machine that is capable of being operated with lower horsepower.

Further, according to the present invention, the time period within which each of the actuators of the electrically-driven hydraulic work machine obtains a predetermined speed can be extended.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a view depicting a hydraulic drive system for an electrically-driven hydraulic work machine according to a first embodiment of the present invention.

FIG. **2** is a view depicting an appearance of an electrically-driven hydraulic excavator in which the hydraulic drive system of the present invention is incorporated.

FIG. **3** is a block diagram depicting functions of a controller.

FIG. **4** is a block diagram depicting functions of a virtual torque calculation section of the controller.

FIG. **5**A is a view depicting a characteristic of a first table of the virtual torque calculation section.

FIG. **5**B is a view depicting a characteristic of a second table of the virtual torque calculation section.

FIG. **6** is a block diagram depicting functions of first and second electric motor rotational speed control sections of the controller.

FIG. **7** is a view depicting a characteristic of a variable horsepower controlling table in the first and second electric motor rotational speed control sections.

FIG. **8** is a view depicting a hydraulic drive system for an electrically-driven hydraulic work machine according to a second embodiment of the present invention.

FIG. **9** is a view depicting a hydraulic drive system for an electrically-driven hydraulic work machine according to a third embodiment of the present invention.

FIG. **10** is a block diagram depicting functions of a controller for a pilot pump.

FIG. **11** is a view depicting a characteristic of a table for calculation of an increase/decrease amount of a virtual displacement of the controller for a pilot pump.

FIG. **12**A is a view depicting a modification to the characteristic of the first table of the virtual torque calculation section.

FIG. **12**B is a view depicting a modification to the characteristic of the second table of the virtual torque calculation section.

FIG. **12**C is a view depicting another modification to the characteristic of the first table of the virtual torque calculation section.

FIG. **12**D is a view depicting another modification to the characteristic of the second table of the virtual torque calculation section.

FIG. **13** is a view depicting a hydraulic drive system for an electrically-driven hydraulic work machine according to a fourth embodiment of the present invention.

FIG. **14** is a block diagram depicting functions of the controller.

FIG. **15** is a block diagram depicting a function of a proportional solenoid valve target current calculation section of the controller.

FIG. **16**A is a view depicting a characteristic of a first table of the proportional solenoid valve target current calculation section.

FIG. **16**B is a view depicting a characteristic of a second table of the proportional solenoid valve target current calculation section.

FIG. **17** is a view depicting a horsepower control characteristic by a regulator piston of a main pump of variable displacement type.

MODES FOR CARRYING OUT THE INVENTION

In the following, embodiments of the present invention are described with reference to the drawings.

First Embodiment

—Structure—

FIG. **1** is a view depicting a hydraulic drive system for an electrically-driven hydraulic work machine (hydraulic excavator) according to a first embodiment of the present invention.

The hydraulic drive system of the present embodiment includes electric motors **101** and **201** (first and second electric motors), main pumps **102** and **202** (first and second hydraulic pumps) of fixed displacement type driven by the electric motors **101** and **201**, respectively, pilot pumps **130** and **230** of fixed displacement type, a boom cylinder **3***a*, an arm cylinder **3***b*, a swing motor **3***c*, a bucket cylinder **3***d*, a swing cylinder **3***e*, travel motors **3***f* and **3***g*, and a blade cylinder **3***h* that are a plurality of actuators driven by hydraulic fluids delivered from the main pumps **102** and **202** of the fixed displacement type, hydraulic fluid supply lines **105** and **205** that introduce hydraulic fluids delivered from the main pumps **102** and **202** of the fixed displacement type to the plurality of actuators **3***a*, **3***b*, **3***c*, **3***d*, **3***e*, **3***f*, **3***g* and **3***h*, and a control valve block **4** that is provided on the downstream of the hydraulic fluid supply lines **105** and **205** and to which hydraulic fluids delivered from the main pumps **102** and **202** of the fixed displacement type are introduced.

The control valve block **4** includes a common hydraulic fluid supply line **305** that is connected to the hydraulic fluid supply lines **105** and **205** and in which hydraulic fluids delivered from the main pumps **102** and **202** are merged, a plurality of flow control valves **6***a*, **6***b*, **6***c*, **6***d*, **6***e*, **6***f*, **6***g*, and **6***h* that are connected to the common hydraulic fluid supply line **305** and control a direction and a flow rate of hydraulic fluid to be supplied from the common hydraulic fluid supply line **305** to each of the plurality of actuators **3***a*, **3***b*, **3***c*, **3***d*, **3***e*, **3***f*, **3***g*, and **3***h*, pressure compensating valves **7***a*, **7***b*, **7***c*, **7***d*, **7***e*, **7***f*, **7***g*, and **7***h* that respectively control differential pressures across the plurality of flow control valves **6***a*, **6***b*, **6***c*, **6***d*, **6***e*, **6***f*, **6***g*, and **6***h*, a main relief valve **14** that is provided in the common hydraulic fluid supply line **305** and controls a pressure of the common hydraulic fluid supply line **305** so as not to become equal to or higher than a set pressure, shuttle valves **9***a*, **9***b*, **9***c*, **9***d*, **9***e*, **9***f*, and **9***g* that are each connected to a load port of each of the plurality of flow control valves **6***a*, **6***b*, **6***c*, **6***d*, **6***e*, **6***f*, **6***g*, and **6***h* and detects a highest load pressure Plmax of the plurality of actuators **3***a*, **3***b*, **3***c*, **3***d*, **3***e*, **3***f*, **3***g*, and **3***h* and outputs the detected highest load pressure Plmax to a highest load hydraulic fluid line **306**, and an unload valve **15** that is connected to the common hydraulic fluid supply line **305** and selects, when the pressure of the common hydraulic fluid supply line **305** becomes higher than a pressure (unload valve set pressure) obtained by adding a set pressure of a spring to the highest load pressure Plmax of the plurality actuators **3***a*, **3***b*, **3***c*, **3***d*, **3***e*, **3***f*, **3***g*, and **3***h*, an open state to return the hydraulic fluid of the common hydraulic fluid supply line **305** to a tank.

The spring that determines an operation pressure of the unload valve **15** has a spring force set such that, when operation levers of a plurality of operation devices that command operation of the plurality of actuators **3***a*, **3***b*, **3***c*, **3***d*, **3***e*, **3***f*, **3***g*, and **3***h* are in a neutral position, the pressure of the common hydraulic fluid supply line **305** becomes a little higher than a target LS differential pressure Pgr (hereinafter described).

Further, the hydraulic drive system of the present embodiment includes a hydraulic fluid supply line **31** to which hydraulic fluids delivered from the pilot pumps **130** and **230** of the fixed displacement type are introduced through check valves **133** and **233**, respectively, a pilot relief valve **32** that is connected to the hydraulic fluid supply line **31** and keeps the pressure of the hydraulic fluid supply line **31** fixed, a gate lock valve **100** that is connected to the hydraulic fluid supply line **31** and selects whether a pilot hydraulic fluid line on the downstream side is to be connected to the hydraulic fluid supply line **31** or connected to the tank, and a gate lock lever **24** disposed on the driver's seat entrance side of the hydraulic work machine for performing a selection operation of the gate lock valve **100**.

The pilot hydraulic fluid line on the downstream side of the gate lock valve **100** is connected to a plurality of pilot valves **60***a*, **60***b*, **60***c*, **60***d*, **60***e*, **60***f*, **60***g*, and **60***h* (refer to FIG. **13**) provided for the plurality of operation devices, and the plurality of pilot valves **60***a*, **60***b*, **60***c*, **60***d*, **60***e*, **60***f*, **60***g*, and **60***h* (refer to FIG. **13**) generate a pilot operation pressure according to an operation amount thereof on the basis of hydraulic fluid of the pilot hydraulic fluid line on the downstream side of the gate lock valve **100** to perform selection control of the plurality of flow control valves **6***a*, **6***b*, **6***c*, **6***d*, **6***e*, **6***f*, **6***g* and **6***h* with the pilot operation pressure.

The hydraulic drive system of the present embodiment further includes inverters **160** and **260** that respectively control the rotational speeds of the electric motors **101** and **201**, step-up/step-down choppers **161** and **261** that respectively supply electric power with a fixed voltage to the inverters **160** and **260**, power storage devices **170** and **270** (first and second power storage devices) connected so as to supply electric power to the electric motors **101** and **201** through the step-up/step-down choppers **161** and **261** and the inverters **160** and **260**, respectively, battery management systems (BMS) **171** and **271** that output information of a voltage, a temperature, and so forth of the power storage devices **170** and **270** to a controller **50** described later, a reference rotational speed instruction dial **56** for indicating a maximum speed of the plurality of actuators **3***a* to **3***h*, a pressure sensor **40** that is provided in the common hydraulic fluid supply line **305** and detects a pressure of the common hydraulic fluid supply line **305**, namely, a delivery pressure (hereinafter referred to suitably as pump pressure) Ps of the main pumps **102** and **202**, another pressure sensor **41** that is provided in the highest load hydraulic fluid line **306** and detects a pressure of the highest load hydraulic fluid line **306** (highest load pressure Plmax of the plurality of actuators **3***a*, **3***b*, **3***c*, **3***d*, **3***e*, **3***f*, **3***g*, and **3***h*), and a controller **50** that receives signals from the battery management systems **171** and **271**, reference rotational speed instruction dial **56**, and pressure sensors **40** and **41** as inputs thereto to generate rotational speed instruction signals to the inverters **160** and **260**.

FIG. **2** depicts an appearance of an electrically-driven hydraulic excavator in which the hydraulic drive system described above is incorporated.

The electrically-driven hydraulic excavator includes a lower travel structure **501**, an upper swing structure **502**, and a front work implement **504** of the swing type. The front work implement **504** is configured from a boom **511**, an arm **512**, and a bucket **513**. The upper swing structure **502** is swingable by rotation of the swing motor **3***c* with respect to the lower travel structure **501**. A swing post **503** is provided at a front portion of the upper swing structure **502**, and the front work implement **504** is attached for upward and downward movement to the swing post **503**. The swing post **503** is rotatable in a horizontal direction with respect to the upper swing structure **502** by extension and contraction of the swing cylinder **3***e*, and the boom **511**, arm **512**, and bucket **513** of the front work implement **504** are rotatable in the upward and downward directions by extension and contraction of the boom cylinder **3***a*, arm cylinder **3***b*, and bucket cylinder **3***d*. A blade **506** that performs upward and downward movement by extension and contraction of the blade cylinder **3***h* is attached to a central frame **505** of the lower travel structure **501**. The lower travel structure **501** travels by driving left and right crawler belts by rotation of the travel motors **3***f* and **3***g*.

A cabin **508** is installed in the upper swing structure **502**, and in the cabin **508**, a driver's seat **521**, operation devices **522** and **523** (in FIG. **2**, only those on the left side are depicted) for the boom, arm, bucket, and swing in which the pilot valves **60***a* to **60***d* are built), an operation device (not depicted) for boom-swing in which the pilot valve **60***e* is built, an operation device (not depicted) for the blade in which the pilot valve **60***h* is built, operation devices **524***a* and **524***b* (in FIG. **2**, only those on the left side are depicted) for travel in which pilot valves **60***f* and **60***g* are built, the gate lock lever **24**, and so forth are provided.

Each of the operation levers of the operation devices **522** and **523** is capable of being operated in an arbitrary direction with reference to cross directions from a neutral position. When the operation lever of the operation device **522** on the left side is operated in the leftward and rightward directions, then the operation device **522** functions as an operation device for swing and the pilot valve **60***c* (refer to FIG. **13**) for swing operates. When the operation lever of the operation device **522** is operated in the forward and backward directions, then the operation device **522** functions as an operation device for the arm and the pilot valve **60***b* (refer to FIG. **13**) for the arm operates. When the operation lever of the operation device **523** on the right side is operated in the forward and backward directions, then the operation device **523** functions as an operation device for the boom and the pilot valve **60***a* for the boom operates. When the operation lever of the operation device **523** is operated in the leftward and rightward directions, then the operation device **523** functions as an operation device for the bucket and the pilot valve **60***d* (refer to FIG. **13**) for the bucket operates.

FIG. **3** depicts a functional block diagram of the controller **50** described above.

The controller **50** includes a virtual limitation torque calculation section **51** for the main pumps **102** and **202** of the fixed displacement type, and a first electric motor rotational speed control section **52** and a second electric motor rotational speed control section **53**.

The virtual limitation torque calculation section **51** receives information (a voltage, a temperature, and so forth) from the battery management systems **171** and **271** as inputs thereto and performs a predetermined calculation process. Outputs of the virtual limitation torque calculation section **51** are inputted to the first electric motor rotational speed control section **52** and the second electric motor rotational speed control section **53** together with outputs from the pressure sensors **40** and **41** and the reference rotational speed instruction dial **56**. The first electric motor rotational speed control section **52** and the second electric motor rotational speed control section **53** perform a predetermined calculation process using the inputs thereto and output results of the processes to the inverters **160** and **260**, respectively.

FIG. **4** depicts a functional block diagram of the virtual limitation torque calculation section **51** in the controller **50**.

The virtual limitation torque calculation section **51** includes first and second SOC estimation sections **51***a* and **51***b*, a differentiation section **51***c*, and first and second tables **51***d* and **51***e*.

The first and second SOC estimation sections **51***a* and **51***b* receive information (a voltage, a temperature and so forth) from the battery management systems **171** and **271** as inputs thereto, respectively, calculate an SOC (State of Charge) indicative of a charge state of the power storage devices **170** and **270**, and outputs SOC**1** and SOC**2** as the SOCs of them, respectively. The SOC**1** and the SOC**2** are differentiated by the differentiation section **51***c* to calculate ΔSOC (=SOC**1**−SOC**2**). ΔSOC is inputted to the first and second tables **51***d* and **51***e*, by which it is converted into a virtual limitation torque T**1** of the main pump **102** and a virtual limitation torque T**2** of the main pump **202** (horsepower control amounts).

FIGS. **5**A and **5**B depict a characteristic of the first and second tables **51***d* and **51***e*.

As depicted in FIG. **5**A, the characteristic of the table **51***d* is set such that, when the difference ΔSOC in a charge state has a positive value, a predetermined maximum value T**1**_max (fixed) is outputted as the virtual limitation torque T**1**. However, when ΔSOC has a negative value, the virtual limitation torque T**1** that decreases as ΔSOC decreases is outputted, and when ΔSOC reaches −ΔSOC_**0** in the proximity of a minimum value −ΔSOC_max, a minimum value T**1**_min is outputted as the virtual limitation torque T**1**.

As depicted in FIG. **5**B, the characteristic of the table **51***e* is set to a reverse characteristic to the characteristic of the table **51***d*. In particular, the characteristic of the table **51***e* is set such that, when ΔSOC has a negative value, a maximum value T2_max is outputted as the virtual limitation torque T2. However, the characteristic of the table 51e is set such that, when ΔSOC has a positive value, the virtual limitation torque T2 that decreases as ΔSOC increases is outputted, and when ΔSOC reaches ΔSOC_0 in the proximity of the maximum value ΔSOC_max, a minimum value T2_min is outputted as the virtual limitation torque T2.

FIG. 6 depicts a functional block diagram of the first and second electric motor rotational speed control sections 52 and 53 of the controller 50. In the following description, a number in ( ) indicates that it is a number in the case of the second electric motor rotational speed control section 53.

Where outputs from the pressure sensors 40 and 41 are represented by Vps and Vplmax, respectively, the outputs Vps and Vplmax are converted into a pressure of the common hydraulic fluid supply line 305, namely, into a pump pressure Ps and a highest load pressure Plmax of the plurality of actuators 3a to 3h by pressure tables 52a (53a) and 52b (53b), and the difference Pls=Ps−Plmax between them is computed by a differentiation section 52c (53c).

On the other hand, when the output from the reference rotational speed instruction dial 56 is represented by Vec, then the output Vec is converted into a reference rotational speed N_0 by a rotational speed table 52d (53d). Further, the reference rotational speed N_0 is converted into a target LS differential pressure Pgr in accordance with a differential pressure table 52e (53e).

Pls and Pgr described above are inputted to a differentiation section 52f (53f), by which the difference ΔP=Pgr−Pls is computed.

ΔP described above is inputted to a table 52g (53g), by which an increase/decrease amount Δq of the virtual displacement is computed.

Δq described above is added to a virtual displacement q1** (q2**) after reflection of horsepower control before one control step stored in a memory and is further limited with a minimum value and a maximum value therefor by a table 52i (53i) to compute a virtual displacement q1* (q2*) before reflection of new horsepower.

On the other hand, the pump pressure Ps described hereinabove and the virtual limitation torque T1 (T2) that is output of the virtual limitation torque calculation section 51 in the controller 50 are inputted to the variable horsepower control table 52r (53r), by which they are converted into and outputted as a limit value q1*limit (q2*limit) for the virtual displacement as first and second allowable values for the absorption horsepower (consumption horsepower) of the main pumps 102 and 202.

A characteristic of the variable horsepower control tables 52r and 53r is depicted in FIG. 7.

A characteristic indicated by a solid line A1 of the variable horsepower control tables 52r and 53r is a characteristic that simulates so-called horsepower control and is such a characteristic that, as the pump pressure Ps becomes high, the limit value q1*limit (q2*limit) for the virtual displacement of the pump pressure Ps decreases.

Further, the characteristic is such a characteristic that, as the virtual limitation torque T1 (T2) decreases, the characteristic simulating horsepower control changes like a broken line B1 or C1 as indicated by an arrow mark in FIG. 7, and consequently, the degree of the limit becomes stronger, and the limit value q1*limit (q2*limit) for the virtual displacement decreases.

A lower one of the virtual displacement q1* (q2*) before reflection of horsepower control described hereinabove and the limit value q1*limit (q2*limit) for the virtual displacement outputted from the variable horsepower control table 52r (53r) described above is selected as the virtual displacement q1** (q2**) after reflection of horsepower control by the minimum value selection section 52s (53s).

Furthermore, the virtual displacement q1** (q2**) is multiplied by the reference rotational speed N_0 by a multiplication section 52j (53j), and a result of the multiplication is outputted as a target flow rate Q1d (Q2d).

If the physical displacement of the main pumps 102 and 202 of the fixed displacement type is represented by qmax1 (qmax2), then the target flow rate Q1d is multiplied by 1/qmax1 (1/qmax2) of a gain 52k (53k) so as to be converted into a target rotational speed N1d (N2d).

Furthermore, the target rotational speed N1d (N2d) is converted into an input Vinv1 (Vinv2) to the inverter 160 (260) and outputted to the inverter 160 (260) by the table 52m (53m).

~Correspondence to Claims~

In the foregoing, the main pumps 102 and 202 of the fixed displacement type are first and second hydraulic pumps, and hydraulic fluids delivered from the first and second hydraulic pumps are merged in the common hydraulic fluid supply line 305 and supplied to the plurality of flow control valves 6a, 6b, 6c, 6c, 6e, 6f, 6g, and 6h and is further supplied to the plurality of actuators 3a, 3b, 3c, 3d, 3e, 3f, 3g, and 3h.

The electric motors 101 and 201 are first and second electric motors that drive the main pumps 102 and 202 (first and second hydraulic pumps), respectively, and the power storage devices 170 and 270 are first and second power storage devices that supply electric power to the electric motors 101 and 201 (first and second electric motors), respectively.

The variable horsepower control table 52r and the minimum value selection section 52s in the electric motor rotational speed control section 52 of the controller 50 cooperates with the pressure sensor 40 to provide a first horsepower control device configured to decrease, when the delivery pressure of the main pump 102 (first hydraulic pump) increases, the delivery flow rate of the main pump 102 (first hydraulic pump) to control the absorption horsepower of the main pump 102 (first hydraulic pump) so as not to exceed the limit value q1*limit (first allowable value) for the virtual displacement. The variable horsepower control table 53r and the minimum value selection section 53s in the electric motor rotational speed control section 53 of the controller 50 cooperate with the pressure sensor 40 to provide a second horsepower control device configured to decrease, when the delivery pressure of the main pump 202 (second hydraulic pump) increases, the delivery flow rate of the main pump 202 (second hydraulic pump) to control the absorption horsepower of the main pump 202 (second hydraulic pump) so as not to exceed the limit value q2*limit (second allowable value) for the virtual displacement.

The virtual limitation torque calculation section 51 of the controller 50 and the variable horsepower control tables 52r and 53r in the electric motor rotational speed control sections 52 and 53 provide a horsepower distribution control section configured to change at least one of the limit values q1*limit and q2*limit (first and second allowable values) of the first and second horsepower control devices such that the charge state of the power storage device 170 (first power storage device) and the charge state of the power storage device 270 (second power storage device) become equal to each other.

Further, in the present embodiment, the main pumps 102 and 202 (first and second hydraulic pumps) are hydraulic pumps of the fixed displacement type, and the first and second horsepower control devices are configured to control the rotational speeds of the main pump **102** (first hydraulic pump) and the main pump **202** (second hydraulic pump) to control the absorption horsepowers of the main pumps **102** and **202** (first and second hydraulic pumps) such that the absorption horsepower of the main pump **102** does not exceed the limit value q**1***limit (first allowable value) and the absorption horsepower of the main pump **202** does not exceed the limit value q**2***limit (second allowable value).

Tables **52***a* to **52***m* in the electric motor rotational speed control section **52** of the controller **50** and the inverter **160** cooperate with the pressure sensors **40** and **41** and the reference rotational speed instruction dial **56** to provide a first flow control section configured to perform, when at least one of the operation devices **522** and **523**, **524***a*, **524***b*, . . . is operated, load sensing control for controlling a delivery flow rate of the main pump **102** (first hydraulic pump) such that the delivery pressure Ps of the main pump **102** (first hydraulic pump) becomes higher by a target differential pressure (target LS differential pressure Pgr) than the highest load pressure Plmax of the plurality of actuators **3***a* to **3***h*. Tables **53***a* to **53***m* in the electric motor rotational speed control section **53** of the controller **50** and the inverter **260** cooperate with the pressure sensors **40** and **41** and the reference rotational speed instruction dial **56** to provide a second flow control section configured to perform, when at least one of the operation devices **522**, **523**, **524***a*, **524***b*, . . . is operated, load sensing control for controlling a delivery flow rate of the main pump **202** (second hydraulic pump) such that the delivery pressure Ps of the main pump **202** (second hydraulic pump) becomes higher by the target differential pressure (target LS differential pressure Pgr) than the highest load pressure Plmax of the plurality of actuators **3***a* to **3***h*.

The first and second flow control sections described above are configured to control the rotational speeds of the main pumps **102** and **202** (first and second hydraulic pumps) respectively to control the delivery flow rate of the main pumps **102** and **202** (first and second hydraulic pumps) such that the delivery pressures of the main pumps **102** and **202** become higher by the target differential pressure than the highest load pressure of the plurality of actuators **3***a* to **3***h*. The first and second horsepower control devices described above are configured to control the delivery flow rates of the main pumps **102** and **202** (first and second hydraulic pumps) respectively, which are controlled by the load sensing control, such that the absorption horsepower of the main pump **102** (first hydraulic pump) does not exceed the limit value q**1***limit (first allowable value) for the virtual displacement and the absorption horsepower of the main pump **202** (second hydraulic pump) does not exceed the limit value q**2***limit (second allowable value) for the virtual displacement.

The battery management system **171** and the first SOC estimation section **51***a* in the virtual limitation torque calculation section **51** of the controller **50** provide a first charge state estimation section configured to estimate a charge state of the power storage device **170** (first power storage device), and the battery management system **271** and the second SOC estimation section **51***b* in the virtual limitation torque calculation section **51** of the controller **50** provide a second power storage stage estimation section configured to estimate a charge state of the power storage device **270** (second power storage device).

The differentiation section **51***c* and the first and second tables **51***d* and **51***e* in the virtual limitation torque calculation section **51** of the controller **50** provide a horsepower control amount calculation section configured to calculate, when the charge state of the power storage device **170** (first power storage device) estimated by the first charge state estimation section described above is lower than the charge state of the power storage device **270** (second power storage device) estimated by the second charge state estimation section described above, the virtual limitation torque T**1** (first horsepower control amount) for decreasing the absorption horsepower of the main pump **102** (first hydraulic pump) and calculate, when the charge state of the power storage device **270** (second power storage device) estimated by the second charge state estimation section described above is lower than the charge state of the power storage device **170** (first power storage device) estimated by the first charge state estimation section described above, the virtual limitation torque T**2** (second horsepower control amount) for decreasing the absorption horsepower of the main pump **202** (second hydraulic pump).

The variable horsepower control table **52***r* in the electric motor rotational speed control section **52** of the controller **50** provides a first allowable value changing section configured to change the limit value q**1***limit (first allowable value) for the virtual displacement of the first horsepower control device described above on the basis of the virtual limitation torque T**1** (first horsepower control amount) calculated by the horsepower control amount calculation section described above. The variable horsepower control table **53***r* that simulates the horsepower control characteristic in the electric motor rotational speed control section **53** of the controller **50** provides a second allowable value changing section configured to change the limit value q**2***limit (second allowable value) for the virtual displacement of the second horsepower control device described above on the basis of the virtual limitation torque T**2** (second horsepower control amount) calculated by the horsepower control amount calculation section described above.

~Operation~

Operation of the present embodiment is described with reference to FIGS. **1** to **7**.

Hydraulic fluids delivered from the pilot pumps **130** and **230** of the fixed displacement type driven by the electric motors **101** and **201** are supplied to the hydraulic fluid supply line **31** through the check valves **133** and **233**, respectively. To the hydraulic fluid supply line **31**, the pilot relief valve **32** is connected, and a pilot primary pressure Ppi**0** is generated in the hydraulic fluid supply line **31**.

Meanwhile, information such as a voltage, a temperature, and so forth of the power storage devices **170** and **270** is introduced to the first and second SOC estimation sections **51***a* and **51***b* of the virtual limitation torque calculation section **51** in the controller **50** through the battery management systems **171** and **271**, and from the charge state SOC**1** of the power storage device **170** and the charge state SOC**2** of the power storage device **270**, virtual limitation torques T**1** and T**2** of the main pumps **102** and **202** of the fixed displacement type are calculated and outputted to the rotational speed control section **52** of the electric motor **101** and the rotational speed control section **53** of the electric motor **201**, respectively.

(a) Case where the SOCs of the power storage device **170** and the power storage device **270** are equal to each other

First, a case is considered in which, in the functional block diagram of the virtual limitation torque calculation section **51** depicted in FIG. **4**, the charge state SOC**1** of the power storage device **170** estimated by the first SOC estimation section **51***a* and the charge state SOC**2** of the power storage device **270** estimated by the second SOC estimation section **51***b* are equal to each other.

Since SOC1=SOC2, ΔSOC computed by the differentiation section **51***c* becomes ΔSOC=SOC1−SOC2=0, and from the characteristics of the first and second tables **51***d* and **51***e* depicted in FIGS. **5**A and **5**B, the virtual limitation torques T**1** and T**2** of the main pumps **102** and **202** become T**1**=T**1**_max and T**2**=T**2**_max, respectively.

The virtual limitation torques T**1** and T**2** that are outputs of the virtual limitation torque calculation section **51** are introduced to the variable horsepower control tables **52***r* and **53***r* of the electric motor rotational speed control sections **52** and **53**, respectively.

<(a-1) Case where all Operation Levers are Neutral>

Since all operation levers (hereinafter referred to simply as operation levers as necessary) of all operation devices **522**, **523**, **524***a*, **524***b*, . . . are neutral, all flow control valves **6***a*, **6***b*, **6***c*, **6***d*, **6***e*, **6***f*, **6***g*, and **6***h* (refer to FIG. **13**) are in a neutral position.

Therefore, the highest load pressure Plmax of the actuators **3***a*, **3***b*, **3***c*, **3***d*, **3***e*, **3***f*, **3***g*, and **3***h* is equal to the tank pressure through the flow control valves **6***a*, **6***b*, **6***c*, **6***d*, **6***e*, **6***f*, **6***g*, and **6***h* and the shuttle valves **9***a*, **9***b*, **9***c*, **9***d*, **9***e*, **9***f*, and **9***g*.

The highest load pressure Plmax is introduced to the unload valve **15** and the pressure sensor **41**.

Although hydraulic fluids delivered from the main pumps **102** and **202** of the fixed displacement type driven by the electric motors **101** and **201** are introduced to the common hydraulic fluid supply line **305**, since all of the plurality of flow control valves **6***a*, **6***b*, **6***c*, **6***d*, **6***e*, **6***f*, **6***g*, and **6***h* are in their neutral position as described hereinabove, the hydraulic fluid is discharged from the unload valve **15** to the tank.

Since Plmax introduced to the unload valve **15** is equal to the tank pressure (estimated that the tank pressure≈0), the pressure of the common hydraulic fluid supply line **305**, namely, the pump pressure Ps, is kept a little higher than Pgr that is a target LS differential pressure by the work of a spring provided in the unload valve **15**.

The pressure (pump pressure) Ps of the common hydraulic fluid supply line **305** is introduced to the pressure sensor **40**.

An output Vps of the pressure sensor **40** for the pump pressure Ps, an output Vplmax of the pressure sensor **41** for the highest load pressure Plmax and an output Vec of the reference rotational speed instruction dial **56** are inputted to the electric motor rotational speed control sections **52** and **53** in the controller **50** in addition to the virtual limitation torques T**1** and T**2** described hereinabove.

Since the electric motor rotational speed control section **52** and the electric motor rotational speed control section **53** operate similarly, the following description is given taking the first electric motor rotational speed control section **52** as an example.

Vps, Vplmax, and Vec described above are converted into Ps, Plmax, and N_**0** by the tables **52***a*, **52***b* and **52***d*, respectively, and from Ps and Plmax, the difference Pls=Ps−Plmax between them is computed by the differentiation section **52***c* (**53***c*).

The difference ΔP between the target LS differential pressure Pgr converted from the reference rotational speed N_**0** by the table **52***e* and Pls described hereinabove is calculated by the differentiation section **52***f*. At this time, in the case where all operation levers are neutral, Ps is kept at a value a little higher than Pgr as described hereinabove, and since Plmax is the tank pressure (estimated that the tank pressure≈0), Pls=Ps−Plmax is kept at a value a little higher than Pgr. Therefore, ΔP becomes ΔP=Pgr−Pls<0, and the increase/decrease amount Δq of the virtual displacement calculated by the table **52***g* has a negative value.

The increase/decrease amount Δq of the virtual displacement is added to the virtual displacement q**1**** after reflection of horsepower control before one control step, and after the resulting sum is limited with the minimum value and the maximum value by the table **52***i*, it becomes a new virtual displacement q**1*** after reflection of horsepower control. Since the increase/decrease amount Δq of the virtual displacement has a negative value, by repeating the control step, the virtual displacement q**1*** before reflection of horsepower control is kept at the minimum value prescribed by the table **52***i*.

On the other hand, since T**1**_max is inputted as the virtual limitation torque T**1** as described above to the variable horsepower control table **52***r*, the limit value q**1***limit for the virtual displacement has a value on the solid line A**1** in FIG. **7**.

As described hereinabove, in the case where all operation levers are neutral, the pump pressure Ps is kept at a value a little higher than Pgr, and if Ps at this time is represented by Pntr, the limit value q**1***limit for the becomes q**1***limit=qmax1 from FIG. **7**.

As described above, since, in the case where all operation levers are neutral, the virtual displacement q**1*** before reflection of horsepower control is kept at the minimum value prescribed by the table **52***i* and the relationship of q**1***<q**1***limit is satisfied, q**1*** is selected from between q**1*** and q**1***limit by the minimum value selection section **52***s* and becomes the virtual displacement q**1**** after reflection of horsepower control.

The virtual displacement q**1**** after reflection of horsepower control is multiplied by the reference rotational speed N_**0** by the multiplication section **52***j* and becomes a target flow rate Q**1***d*, and the target flow rate Q**1***d* is converted into a target rotational speed N**1***d* by multiplication by 1/qmax1 with the gain **52***k* and is further converted into an output Vnv**1** to the inverter **160** by the table **52***m*.

As described above, since, in the case where all operation levers are neutral, the virtual displacement q**1**** after reflection of horsepower control is kept at the minimum value prescribed by the table **52***i*, also the target flow rate Q**1***d* and the target rotational speed N**1***d* are also kept at minimum values.

In particular, in the case where all operation levers are neutral, the rotational speed of the electric motor **101** is kept at a minimum rotational speed, and also the delivery flow rate from the main pump **102** of the fixed displacement type is kept at its minimum.

Since also the electric motor rotational speed control section **53** operates similarly, the rotational speed of the electric motor **201** is kept at its minimum rotational speed, and also the delivery flow rate from the main pump **202** of the fixed displacement type is kept at a minimum.

<(a-2) Case where Boom Raising is Operated>

A case is considered in which the operation lever of the operation device **523** for boom is inputted in a raising direction, namely, in a direction in which the boom cylinder **3***a* is extended.

The flow control valve **6***a* for the boom selects the leftward direction in the figure, and hydraulic fluids of the hydraulic fluid supply lines **105** and **205** from the main pumps **102** and **202** of the fixed displacement type are supplied to the bottom side of the boom cylinder **3***a* through the common hydraulic fluid supply line **305**, pressure compensating valve **7***a*, and flow control valve **6***a*.

On the other hand, the load pressure of the boom cylinder **3**a is introduced as the highest load pressure Plmax to the pressure compensating valves **7**a, **7**b, **7**c, **7**d, **7**e, **7**f, **7**g, and **7**h, unload valve **15**, and pressure sensor **41** through the load port of the flow control valve **6**a, shuttle valves **9**a, **9**b, **9**c, **9**d, **9**e, **9**f, and **9**g, and highest load hydraulic fluid line **306**.

By the load pressure of the boom cylinder **3**a introduced to the unload valve **15**, the set pressure of the unload valve **15** rises to a pressure of the sum of the spring force of the unload valve **15** and the highest load pressure Plmax (load pressure of the boom cylinder **3**a), whereupon the hydraulic line for discharging the hydraulic fluid in the common hydraulic fluid supply line **305** to the tank is interrupted.

The pressure (pump pressure Ps) of the common hydraulic fluid supply line **305** is introduced to the pressure sensor **40**.

The output Vps of the pressure sensor **40** for the pump pressure Ps, the output Vplmax of the pressure sensor **41** for the highest load pressure Plmax, and the output Vec of the reference rotational speed instruction dial **56** are inputted to the electric motor rotational speed control sections **52** and **53** in the controller **50** in addition to the virtual limitation torques T**1** and T**2** described hereinabove.

Since the electric motor rotational speed control section **52** and the electric motor rotational speed control section **53** operate similarly, the following description is given taking the electric motor rotational speed control section **52** as an example.

Vps, Vplmax, and Vec described hereinabove are converted into Ps, Plmax, and N_**0** by the tables **52**a, **52**b, and **52**d, respectively.

Immediately after boom raising activation, the pump pressure Ps has a value a little higher than the target LS differential pressure similarly as in the case where all operation levers are neutral, and usually, in the case where the boom cylinder **3**a is extended, the load pressure of the boom cylinder **3**a is often higher than the pump pressure. Since, in this case, $Ps<Plmax$ is satisfied, Pls becomes $Pls=Ps-Plmax<0$.

The difference ΔP between the target LS differential pressure Pgr converted from the reference rotational speed N_**0** by the table **52**e and Pls described above is calculated by the differentiation section **52**f.

Since Pls has a negative value as described above, $\Delta P=Pgr-Pls$ has a positive value higher than Pgr.

Although ΔP is converted into an increase/decrease amount Δq of the virtual displacement by the table **52**g, since ΔP becomes a positive value higher than Pgr, also the increase/decrease amount Δq of the virtual displacement has a positive value.

The increase/decrease amount Δq of the virtual displacement is added to the virtual displacement q**1**** after reflection of horsepower control before one control step and is limited with the minimum value and the maximum value by the table **52**i, whereafter it becomes a new virtual displacement q**1*** before reflection of horsepower control.

Since Δq that is the increase/decrease amount of the virtual displacement has a positive value as described above, by repeating the control step, the virtual displacement q**1*** before reflection of horsepower control gradually increases within the range between the minimum value and the maximum value of the table **52**i, and the increase continues until after $\Delta P=Pgr-Pls$ becomes 0, namely, until after Pls becomes equal to Pgr that is the target LS differential pressure.

On the other hand, since T**1**_max is inputted as the virtual limitation torque T**1** as described hereinabove to the variable horsepower control table **52**r, the limit value q**1***limit of the virtual displacement has a value on the solid line A**1** of FIG. **7**.

In the case where, in boom raising operation, the LS differential pressure Pls is equal to the target LS differential pressure Pgr as described hereinabove, the pump pressure Ps is kept at a pressure higher by the target LS differential pressure Pgr than the highest load pressure Plmax.

From FIG. **7**, if the pump pressure Ps upon boom raising operation at this time is represented by Pbmr, then the limit value q**1***limit of the virtual displacement becomes qbmr1 that is lower than qmax simulating horsepower control.

Since the minimum value selection section **52**s selects a smaller one of the virtual displacement q**1*** before reflection of horsepower control described above and the output q**1***limit of the variable horsepower control table **52**r as the virtual displacement q**1**** after reflection of horsepower control, the virtual displacement q**1**** after reflection of horsepower control performs operation that simulates a main pump of variable displacement type that delivers a necessary flow rate within the range of the horsepower control characteristic prescribed by the variable horsepower control table **52**r.

The virtual displacement q**1**** after reflection of horsepower control is multiplied by the reference rotational speed N_**0** by the multiplication section **52**j and becomes a target flow rate Q**1**d, and the target flow rate Q**1**d is converted into a target rotational speed N**1**d by multiplication by 1/qmax1 by the gain **52**k and is further converted into an output Vinv1 to the inverter **160** by the table **52**m.

Also the electric motor rotational speed control section **53** operates similarly, and the flow rates of hydraulic fluids delivered from the main pumps **102** and **202** of the fixed displacement type are merged and supplied to the flow control valve **6**a.

As described hereinabove, the virtual displacements q**1**** and q**2**** after reflection of horsepower control operate simulating a main pump of the variable displacement type that delivers a flow rate necessary within the range of the horsepower control characteristics prescribed by the variable horsepower control tables **52**r and **53**r to control the rotational speeds of the electric motors **101** and **201** such that the flow rate is implemented. Therefore, the virtual displacements q**1**** and q**2**** after reflection of horsepower control operate so as to control the rotational speed of the electric motor **101** such that the total delivery amount of the main pumps **102** and **202** of the fixed displacement type becomes equal to a flow rate required by the flow control valve **6**a within a range within which the consumption horsepower of the main pumps **102** and **202** of the fixed displacement type does not exceed a certain value.

(b) Case where the SOC of the Power Storage Device **170** is Greater than that of the Power Storage Device **270**

A case is considered in which, in the functional block diagram of the virtual limitation torque calculation section **51** depicted in FIG. **4**, the relationship between the charge state SOC**1** of the power storage device **170** estimated by the first SOC estimation section **51**a and the charge state SOC**2** of the power storage device **270** estimated by the second SOC estimation section **51**b is SOC**1**>SOC**2**.

Since SOC**1**>SOC**2**, ΔSOC computed by the differentiation section **51**c becomes $\Delta SOC=SOC1-SOC2>0$, namely, has a positive value, from the characteristics of the tables **51**d and **51**e depicted in FIGS. **5**A and **5**B, the virtual limitation torque T**1** of the main pump **102** of the fixed displacement type becomes T**1**=T**1**_max, and the virtual limitation torque T**2** of the main pump **202** of the fixed displacement type becomes a value lower than T2_max. Here, a case in which ΔSOC=SOC1−SOC2=ΔSOC_0 is considered. From FIG. 5B, T2 at this time is T2=T2_min.

The virtual limitation torques T1 and T2 that are outputs of the virtual limitation torque calculation section **51** are introduced to the variable horsepower control tables **52***r* and **53***r* of the electric motor rotational speed control sections **52** and **53**, respectively.

<(b-1) Case where all Operation Levers are Neutral>

In the case where all operation levers are neutral, both of the main pumps **102** and **202** of the fixed displacement type deliver a minimum flow rate at a minimum rotational speed similarly as in the case of (a) SOC1=SOC2 described hereinabove.

<(b-2) Case where Boom Raising Operation is Performed>

Basic operation in which the main pumps **102** and **202** deliver a necessary flow rate by load sensing control within ranges of the horsepower prescribed by the virtual limitation torques T1 and T2, respectively, is similar to that in the case of (a) SOC1=SOC2 described hereinabove.

In the case where SOC1>SOC2 and ΔSOC (=SOC1−SOC2) is ΔSOC_0, T1=T1_max and T2=T2_min as described hereinabove.

Since the virtual limitation torque T1 of the main pump **102** of the fixed displacement type is T1=T1_max, operation quite same as that of (a) described above is performed.

On the other hand, the virtual limitation torque T2 of the main pump **202** of the fixed displacement type is T2=T2_min and is lower than the virtual limitation torque T1 of the main pump **102**, and as indicted by a broken line C1 in FIG. **7**, the virtual displacement q2** after reflection of horsepower control is limited to qbmr2_min by the table **53***r*.

The virtual displacement q2** after reflection of horsepower control is multiplied by the reference rotational speed N_0 by the multiplication section **53***j* and becomes a target flow rate Q2*d*, and the target flow rate Q2*d* is converted into a target rotational speed N2*d* by multiplication by 1/qmax2 with the gain **53***k* and is further converted into an output Vinv2 to the inverter **260** by the table **53***m*.

At this time, since the virtual displacement q2** after reflection of horsepower control of the main pump **202** of the fixed displacement type is limited to qbmr2_min as described hereinabove, the rotational speed N2*d* is limited so as to be lower than the rotational speed N1*d* of the main pump **102** of the fixed displacement type.

In this manner, in the case where SOC1>SOC2, namely, in the case where the SOC of the power storage device **270** is smaller than the SOC of the power storage device **170**, the flow rate delivered from the main pump **202** that is driven by electric power supplied from the power storage device **270** is lower than the flow rate delivered from the main pump **102** that is driven by electric power supplied from the power storage device **170**.

Although the power consumption of the pump increases in proportion to the pressure×flow rate and the pressure (pump pressure Ps) of the common hydraulic fluid supply line **305** is common and equal, since the flow rate of the main pump **202** is lower than the flow rate of the main pump **102** as described above, the power consumption of the main pump **202** becomes smaller than the power consumption of the main pump **102**.

Therefore, the electric power consumption of the power storage device **270** from which electric power is supplied to the main pump **202** becomes lower than the electric power consumption of the power storage device **170** from which electric power is supplied to the main pump **102**.

Since the electric power consumption of the power storage device **270** from which electric power is supplied to the main pump **202** is lower than the electric power consumption of the power storage device **170** from which electric power is supplied to the main pump **102**, the rate at which the SOC2 of the power storage device **270** decreases becomes lower than the rate at which the SOC1 of the power storage device **170** decreases, and this continues until after the SOC1 becomes equal to the SOC2.

If SOC1=SOC2 is satisfied, then operation same as that in the case of (a) is performed.

(c) Case where the SOC of the Power Storage Device **270** is Higher than that of the Power Storage Device **170**

A case is considered in which, in the functional block diagram of the virtual limitation torque calculation section **51** depicted in FIG. **4**, the relationship between the charge state SOC1 of the power storage device **170** estimated by the first SOC estimation section **51***a* and the charge state SOC2 of the power storage device **270** estimated by the second SOC estimation section **51***b* is SOC1<SOC2 is considered.

In the following, the relationship between the main pumps **102** and **202** of the fixed displacement type is reverse to that in the case of SOC1>SOC2 of (b).

Since SOC1<SOC2, ΔSOC computed by the differentiation section **51***c* becomes ΔSOC=SOC1−SOC2<0, namely, has a negative value, from the characteristic of the tables **51***d* and **51***e* depicted in FIGS. **5**A and **5**B, the virtual limitation torque T2 of the main pump **202** of the fixed displacement type becomes T2=T2_max and the virtual limitation torque T1 of the main pump **102** of the fixed displacement type has a value lower than T1_max. Here, a case in which ΔSOC=SOC1−SOC2=−ΔSOC_0 is considered. From FIG. 5A, the virtual limitation torque T1 at this time is T1=T1_min.

The virtual limitation torques T1 and T2 that are outputs of the virtual limitation torque calculation section **51** are introduced to the variable horsepower control tables **52***r* and **53***r* of the electric motor rotational speed control sections **52** and **53**, respectively.

<(c-1) Case where all Operation Levers are Neutral>

In the case where all operation levers are neutral, both of the main pumps **102** and **202** deliver a minimum flow rate at a minimum rotational speed similarly as in the cases of (a) SOC1=SOC2 and (b) SOC1>SOC2 described above.

<(c-2) Case where Boom Raising is Operated>

Basic operation in which the main pumps **102** and **202** deliver hydraulic fluids of necessary flow rates by load sensing control within ranges of the horsepower prescribed by the virtual limitation torques T1 and T2, respectively, is similar to that in the case of (a) SOC1=SOC2 described hereinabove.

In the case where SOC1<SOC2 and ΔSOC (=SOC1−SOC2) is −ΔSOC_0 as described above, T1=T1_min and T2=T2_max.

Since the virtual limitation torque T2 of the main pump **202** of the fixed displacement type is T2=T2_max, operation quite same as that in (a) described hereinabove is performed.

On the other hand, the virtual limitation torque T1 of the main pump **102** of the fixed displacement type is T1=T1_min and is lower than the virtual limitation torque T2 of the main pump **202**, and as indicated by the broken line C1 of FIG. **7**, the virtual displacement q1** after reflection of horsepower control is limited to qbmr1_min by the table **52***r*.

The virtual displacement q1** after reflection of horsepower control is multiplied by the reference rotational speed N_0 by the multiplication section **52***j* and becomes a target flow rate Q1*d*, and the target flow rate Q1*d* is converted into a target rotational speed N1*d* by multiplication by 1/qmax1 by the gain **52***k* and is further converted into an output Vinv1 to the inverter **160** by the table **52***m*.

At this time, since the virtual displacement q1** after reflection of horsepower control of the main pump **102** of the fixed displacement type is limited to qbmr1_min as described above, the rotational speed N1*d* is limited so as to be lower than the rotational speed N2*d* of the main pump **202** of the fixed displacement type.

In this manner, in the case where SOC1<SOC2, namely, in the case where the SOC of the power storage device **170** is lower than the SOC of the power storage device **270**, the flow rate of hydraulic fluid delivered from the main pump **102** that is driven by electric power supplied from the power storage device **170** is lower than the flow rate of hydraulic fluid delivered from the main pump **202** that is driven by electric power supplied from the power storage device **270**.

Although the power consumption of the pump increases in proportion to the pressure×the flow rate and the pressure (pump pressure Ps) of the common hydraulic fluid supply line **305** is common and equal, since the flow rate of the main pump **102** is lower than the flow rate of the main pump **202** as described above, the power consumption of the main pump **102** becomes lower than the power consumption of the main pump **202**.

Therefore, the electric power consumption of the power storage device **170** from which electric power is supplied to the main pump **102** becomes lower than the electric power consumption of the power storage device **270** from which electric power is supplied to the main pump **202**.

Since the electric power consumption of the power storage device **170** from which electric power is supplied to the main pump **102** is lower than the electric power consumption of the power storage device **270** from which electric power is supplied to the main pump **202**, the rate at which the SOC1 of the power storage device **170** decreases becomes lower than the rate at which the SOC2 of the power storage device **270** decreases, and this continues until after the SOC1 and the SOC2 become equal to each other.

If SOC1=SOC2 is satisfied, then operation same as that in the case of (a) is performed.

~Advantage~

According to the present embodiment, the following advantages are obtained.

Since the hydraulic drive system is configured such that it includes, in addition to the main pump **102**, electric motor **101**, and power storage device **170**, the main pump **202**, electric motor **201**, power storage device **270**, and the common hydraulic fluid supply line **305** in which hydraulic fluids delivered from the main pumps **102** and **202** are merged and the merged hydraulic fluid is supplied to the plurality of flow control valves **6***a*, **6***b*, **6***c*, **6***d*, **6***e*, **6***f*, **6***g*, and **6***h* and further supplied to the plurality of actuators **3***a*, **3***b*, **3***c*, **3***d*, **3***e*, **3***f*, **3***g*, and **3***h*, the rated voltages of various electric equipment such as a power storage device can be made common to that of an electrically-driven hydraulic work machine that requires lower horsepower.

Further, the hydraulic drive system includes the virtual limitation torque calculation section **51** and the variable horsepower control tables **52***r* and **53***r* and, in the case where the charge state of one of the power storage devices **170** and **270** becomes lower than the charge state of the other one of the power storage devices **170** and **270**, the limit value q1*limit (q2*limit) of the virtual displacement is changed such that the charge state of the power storage device **170** and the charge state of the power storage device **270** become equal to each other thereby to suppress the power consumption of the hydraulic pump whose charge state is lower. Therefore, even in the case where there is a difference between the machine efficiencies of the main pumps **102** and **202** or between the efficiencies of such electric equipment as the inverters **160** and **260** or the step-up/step-down choppers **161** and **261**, or even in the case in which there is a difference between the electric power consumption amounts or the charge state characteristics of the power storage devices **170** and **270**, the difference gradually decreases while the charge states of the power storage devices **170** and **270** are controlled so as to become equal to each other. Therefore, it is prevented that the power storage situation of only one of the power storage devices **170** and **270** decreases significantly, and the period of time within which the actuators of the electrically-driven hydraulic work machine obtain a predetermined speed.

It is to be noted that, while, in the embodiment described above, the two tables of the first and second tables **51***d* and **51***e* and the two motor rotational speed control sections of the electric motor rotational speed control sections **52** and **53** are provided in the virtual limitation torque calculation section **51** of the controller **50** and the limit value q1*limit or the limit value q2*limit (first or second allowable value) is changed such that, even in the case where the charge state of one of the two power storage devices **170** and **270** becomes lower than the charge state of the other power storage device, the charge state of the power storage device **170** and the charge state of the power storage device **270** become equal to each other, only one of the first and second tables **51***d* and **51***e* (for example, the first table **51***d*) and a corresponding one of the electric motor rotational speed control sections **52** and **53** (for example, the electric motor rotational speed control section **52**) may be provided and the limit value q1*limit (first allowable value) may be changed such that, only in the case where the charge state of one (for example, the power storage device **270**) of the two power storage devices **170** and **270** becomes lower than the charge state of the other power storage device (for example, the power storage device **170**), the charge state of the power storage device **170** and the power storage device of the power storage device **270** become equal to each other. Also this makes it possible to prevent that the charge state of one (for example, the power storage device **270**) of the power storage devices **170** and **270** becomes significantly lower than the charge state of the other one (for example, the power storage device **170**), and the period of time within which the actuators of the electrically-driven hydraulic work machine can obtain a predetermined speed can be extended.

~Modifications to the Characteristic of the First and Second Tables **51***d* and **51***e*~

FIGS. **12**A and **12**B depict a first modification to the characteristic of the tables **51***d* and **51***e*.

While, in FIGS. **5**A and **5**B, the characteristic of the tables **51***d* and **51***e* is set such that a fixed value (maximum value T1_max) is outputted as the virtual limitation torque T1 when ΔSOC has a positive value (FIG. **5**A) or when ΔSOC has a negative value (FIG. **5**B), the virtual limitation torque T1 may otherwise be set such that, as ΔSOC increases (FIG. **12**A) or decreases (FIG. **12**B), the virtual limitation torque T1 further increases from a value equal to the maximum value T1_max to a value T1_od1. Consequently, in the case where the power storage device **170** and the power storage device **270** are different in SOC from each other, since not only the electric power consumption of the power storage device with a lower SOC decreases but also the electric power consumption of the power storage device with a greater SOC increases, the SOC1 of the power storage device **170** and the SOC2 of the power storage device **270** can be made equal to each other in a shorter period of time.

A second modification to the characteristics of the tables **51***d* and **51***e* is depicted in FIGS. **12**C and **12**D.

While, in FIGS. **12**A and **12**B, the virtual limitation torque T**1** is set such that, as ΔSOC increases from 0 (FIG. **12**A) or decreases (FIG. **12**B), the virtual limitation torque T**1** further increases from a value equal to the maximum value T**1**_max to the value T**1**_od**1**, in FIGS. **12**C and **12**D, the virtual limitation torque T**1** is set such that a dead zone of 0 to ΔSOC_**1** (FIG. **12**C) or 0 to −ΔSOC_**1** (FIG. **12**D) is provided and the virtual limitation torque T**1** further increases from a value equal to the maximum value T**1**_max to the value T**1**_od**1** when ΔSOC goes out of the dead zone. This increases the electric power consumption of the power storage device with a higher SOC only in the case where ΔSOC goes out of the dead zone, and the stability of control can be achieved.

Second Embodiment

~Structure~

FIG. **8** is a view depicting a hydraulic drive system of an electrically-driven hydraulic work machine (hydraulic excavator) according to a second embodiment of the present invention.

The hydraulic drive system of the present embodiment further includes, in addition to the configuration of the first embodiment depicted in FIG. **1**, a check valve **180** (first check valve) that is provided in the hydraulic fluid supply line **105** of the main pump **102** of the fixed displacement type and blocks a flow of hydraulic fluid from the common hydraulic fluid supply line **305** to the main pump **102** of the fixed displacement type, and a check valve **280** (second check valve) that is provided in the hydraulic fluid supply line **205** of the main pump **202** of the fixed displacement type and blocks a flow of hydraulic fluid from the common hydraulic fluid supply line **305** to the main pump **202** of the variable displacement type.

Further, the hydraulic drive system of the present embodiment includes an inputting device **58** and is configured such that, in such a case where the SOC of one of the power storage devices **170** and **270** decreases significantly in comparison with the other SOC or in such a case where only one of the power storage devices **170** and **270** is used and the electric power stored in the other power storage device is preserved while the total operating time period is to be increased in place of suppressing the work amount of the hydraulic work machine to be low, when an operator operates the inputting device **58**, then the controller **50** stops one of the electric motors **101** and **201** for driving the main pumps **102** and **202** of the fixed displacement type.

The structure of the other part is same as that of the first embodiment.

~Operation~

In the present embodiment configured in such a manner as described above, since the check valves **180** and **280** are provided in the hydraulic fluid supply lines **105** and **205**, respectively, in the case where one of the electric motors **101** and **201** that respectively drive the main pumps **102** and **202** of the fixed displacement type is stopped, the pressure of the common hydraulic fluid supply line **305** is prevented from being applied to the delivery port of one of the main pumps **102** and **202** of the fixed displacement type which is being stopped.

Operation of the other part is same as that in the first embodiment.

~Advantage~

According to the present embodiment, the following advantage is achieved in addition to the advantages of the first embodiment.

In the case where the operator operates the inputting device **58** to stop one of the electric motors **101** and **201** for driving the main pumps **102** and **202** of the fixed displacement type, the check valves **180** and **280** prevent hydraulic fluid from leaking out from the main pump of the fixed displacement type in the stopping state to the tank, and power of the hydraulic fluid delivered from the main pump of the fixed displacement type in the driven state can be prevented from being lost wastefully by leak of the hydraulic fluid.

Third Embodiment

~Structure~

FIG. **9** is a view depicting a hydraulic drive system of an electrically-driven hydraulic work machine (hydraulic excavator) according to a third embodiment of the present invention.

The hydraulic drive system of the present embodiment is configured such that it does not include the pilot pumps **130** and **230** of the fixed displacement type that are provided in the first embodiment and are driven by the electric motors **101** and **201** and the electric motors **101** and **201** drive only the main pumps **102** and **202** of the fixed displacement type.

Further, the hydraulic drive system of the present embodiment includes an electric motor **301** (third electric motor), a pilot pump **30** of fixed displacement type that is driven by the electric motor **301**, an inverter **360** that controls the rotational speed of the electric motor **301**, a step-up/step-down chopper **361** that supplies electric power with a fixed voltage to the inverter **360**, a pressure sensor **42** that is provided in the hydraulic fluid supply line **31** of the pilot pump **30** and detects a pressure of the hydraulic fluid supply line **31**, namely, a delivery pressure (hereinafter referred to suitably as pump pressure) of the pilot pump **30**, and a controller **55** that generates a rotational speed instruction signal to the inverter **360** on the basis of a detection signal of the pressure sensor **42** to control the inverter **360**.

Similarly as in the first embodiment, the power storage device **270** is configured so as to supply electric power to the electric motor **201** through the step-up/step-down chopper **261** and the inverter **260** and supply electric power also to the electric motor **301** through the step-up/step-down chopper **361** and the inverter **360**.

Hydraulic fluid delivered from the pilot pump **30** of the fixed displacement type is supplied to the hydraulic fluid supply line **31**. To the hydraulic fluid supply line **31**, a gate lock valve **100** is connected which selects the pilot hydraulic fluid line on the downstream side so as to be connected to the hydraulic fluid supply line **31** or to the tank, and for the gate lock valve **100**, a gate lock lever **24** is provided.

FIG. **10** depicts a functional block diagram of the controller **55** in the third embodiment.

The controller **55** includes a table **55***a*, another table **55***b*, an addition section **55***c*, a further table **55***d*, and a still further table **55***f*.

If the output from the pressure sensor **42** is represented by Vpi, then the output Vpi is converted into a pressure (pilot primary pressure Ppi) of the hydraulic fluid supply line **31** by the table **55***a* and is converted into an increase/decrease amount Δqi of the virtual displacement qi of the pilot pump **30** by the table **55***b*.

FIG. **11** depicts a characteristic of the table **55***b*.

The characteristic of the table **55***b* is configured such that, where the pilot primary pressure Ppi of a target is represented by Ppi**0**, when the pressure Ppi of the hydraulic fluid supply line **31** is lower than Ppi**0**, a positive value from 0 to Δqi_max is outputted as Δqi; when Ppi is higher than Ppi**0**, a negative value between Δqi_min and 0 is outputted as Δqi; and when Ppi is equal to Ppi**0**, Δqi=0 is outputted.

The virtual displacement increase/decrease amount Δqi of the pilot pump is added to a pilot pump virtual displacement qi before one control state by the addition section **55***c* to make a new pilot pump virtual displacement qi.

The pilot pump virtual displacement qi is configured so as to be converted into a target rotational speed Npi of the electric motor **301** by the table **55***d*, and further converted into an input Vinv**3** to the inverter **360** by the table **55***f*, and then outputted to the inverter **360**.

The structure of the other part is same as that of the first embodiment.

~Correspondence to Claims~

In the foregoing, the electric motor **301** is a third electric motor to which electric power is supplied by one of the power storage devices **170** and **270** (first and second power storage devices) to drive the pilot pump **30**, and the controller **55** cooperates with the pressure sensor **42** and the inverter **360** to provide a pilot pump control device configured to control the rotational speed of the electric motor **301** (third electric motor) such that the pilot primary pressure Ppi generated by the pilot pump **30** becomes equal to a target pressure (target pilot primary pressure Ppi**0**).

~Operation~

Operation of the third embodiment is described below with reference to FIGS. **9**, **10**, and **11**.

The rotational speed controlling function of the electric motors **101** and **201** for driving the main pumps **102** and **202** of the fixed displacement type is same as that of the first embodiment.

Also the function of controlling the virtual limitation torques T**1** and T**2** of the main pumps **102** and **202** of the fixed displacement type depending upon the charge states of the power storage devices **170** and **270**, respectively, to control the charge states of the power storage devices **170** and **270** so as to eliminate an imbalance between the charge states.

The third embodiment is different from the first embodiment in that the pilot pump **30** of the fixed displacement type is driven independently by the electric motor **301** different from the electric motors **101** and **201** that respectively drive the main pumps **102** and **202** of the fixed displacement type.

In the following, operation of the present embodiment in rotational speed control of the electric motor **301** for driving the pilot pump **30** is described.

(a) Case where the Pressure of the Hydraulic Fluid Supply Line **31** is Lower than the Target Pilot Primary Pressure

A case is considered in which the pressure (pilot primary pressure) of the hydraulic fluid supply line **31** is lower than the pilot primary pressure Ppi**0**.

As depicted in FIG. **10**, Vpi inputted by the pressure sensor **42** is converted into a pilot primary pressure Ppi by the table **55***a*.

In the case where Ppi<Ppi**0**, from FIG. **11**, the pilot pump virtual displacement increase/decrease amount Δqi has a positive value from 0 to Δqi_max.

The pilot pump virtual displacement increase/decrease amount Δqi is added to a pilot pump virtual displacement qi before one control step, and in the case where Ppi is lower than the target pilot primary pressure Ppi**0** as described above, the virtual displacement qi gradually increases.

The increase continues until after the pilot primary pressure Ppi becomes equal to the pilot primary pressure Ppi**0**.

The pilot pump virtual displacement qi is converted into a target rotational speed Npi by the table **55***d* and into an output Vinv**3** to the inverter **360** by the table **55***f* to control the rotational speed of the electric motor **301**.

In particular, in the case where the pressure of the hydraulic fluid supply line **31** is lower than the target pilot primary pressure Ppi**0**, the electric motor **301** increases its rotational speed until after the pressure (pilot primary pressure) of the hydraulic fluid supply line **31** becomes equal to the target pilot primary pressure.

(b) Case where the Pressure of the Hydraulic Fluid Supply Line **31** is Higher than the Target Pilot Primary Pressure

A case is considered in which the pressure (pilot primary pressure) of the hydraulic fluid supply line **31** is higher than the target pilot primary pressure Ppi**0**.

As depicted in FIG. **10**, Vpi inputted by the pressure sensor **42** is converted into a pilot primary pressure Ppi by the table **55***a*.

In the case where Ppi>Ppi**0**, from FIG. **11**, the pilot pump virtual displacement increase/decrease amount Δqi has a negative value between Δqi_min and 0.

The pilot pump virtual displacement increase/decrease amount Δqi is added to the pilot pump virtual displacement qi before one control step, and in the case where Ppi is higher than the pilot primary pressure Ppi**0**, the pilot pump virtual displacement qi gradually decreases as described hereinabove.

This decrease continues until after the pilot primary pressure Ppi becomes equal to the target pilot primary pressure Ppi**0**.

The pilot pump virtual displacement qi is converted into a target rotational speed Npi by the table **55***d* and into an output Vinv**3** to the inverter **360** by the table **55***f* to control the rotational speed of the electric motor **301**.

In particular, in the case where the pressure of the hydraulic fluid supply line **31** is higher than the target pilot primary pressure Ppi**0**, the electric motor **301** decreases its rotational speed until after the pressure (pilot primary pressure) of the hydraulic fluid supply line **31** becomes equal to the target pilot primary pressure.

As described in (a) and (b) above, the controller **55** controls the rotational speed of the electric motor **301** such that the pressure (pilot primary pressure) of the hydraulic fluid supply line **31** becomes equal to the target pilot primary pressure (Ppi**0**).

~Advantage~

According to the present embodiment, the following advantages are achieved in addition to the advantages of the first embodiment.

In the present embodiment, since the electric power consumed by the power storage device **270** is equal to the sum of the electric power consumption of the electric motor **201** for driving the main pump **202** of the fixed displacement type and the electric power consumption of the electric motor **301** for driving the pilot pump of the fixed displacement type, in comparison with the case of the first embodiment, the charge state of the power storage device **270** tends to decrease earlier than the charge state of the power storage device **170**. However, similarly as in the first embodiment, in the case where the charge state of one of the power storage device (in the third embodiment, the power storage device **270**) becomes lower than the charge state of the other power storage device (in the third embodiment, the power storage device **170**), the rotational speed of the hydraulic pump (in the third embodiment, **202**) that is driven by the electric motor to which electric power is supplied from the power storage device in the lower charge state (in the third embodiment, the power storage device **270**) is limited and the electric power consumption of the electric motor is suppressed, and the charge states of the two power storage devices are controlled so as to become equal to each other again.

Therefore, similarly as in the first embodiment, since the charge state of one of the two power storage devices **170** and **270** is prevented from extremely unevenly being decreased, it can be prevented that the period of time within which the hydraulic work machine operates using the two power storage devices **170** and **270** decreases.

Fourth Embodiment

~Structure~

FIG. **13** is a view depicting a hydraulic drive system of an electrically-driven hydraulic work machine (hydraulic excavator) according to a fourth embodiment of the present invention.

The hydraulic drive system of the present embodiment includes electric motors **101**, **201**, **401**, and **601**, main pumps **122**, **222**, **422** and **622** of the variable displacement type driven by the electric motors **101**, **201**, **401**, and **601**, respectively, an electric motor **301**, a pilot pump **30** of the fixed displacement type driven by the electric motor **301**, a boom cylinder **3***a*, an arm cylinder **3***b*, a swing motor **3***c*, a bucket cylinder **3***d*, a swing cylinder **3***e*, travel motors **3***f* and **3***g*, and a blade cylinder **3***h* that are a plurality of actuators driven by hydraulic fluids delivered from the main pumps **122**, **222**, **422**, and **622** of the variable displacement type, hydraulic fluid supply lines **105***a*, **105***b*, **205***a*, and **205***b* for introducing hydraulic fluids delivered from the main pumps **122**, **222**, **422**, and **622** of the variable displacement type into the plurality of actuators **3***a*, **3***b*, **3***c*, **3***d*, **3***e*, **3***f*, **3***g*, and **3***h*, a common hydraulic fluid supply line **307** that is connected to the hydraulic fluid supply lines **105***a* and **105***b* and in which hydraulic fluids delivered from the main pumps **122** and **222** (first and second hydraulic pumps) are merged, a common hydraulic fluid supply line **308** that is connected to the hydraulic fluid supply lines **205***a* and **205***b* and in which hydraulic fluids delivered from the main pumps **422** and **622** (first and second hydraulic fluid pumps) are merged, and a control valve block **104** that is provided on the downstream of the common hydraulic fluid supply lines **307** and **308** and to which hydraulic fluids delivered from the main pumps **122** and **222** and merged in the common hydraulic fluid supply line **307** and hydraulic fluids delivered from the main pumps **422** and **622** and merged in the common hydraulic fluid supply line **308** are introduced.

The control valve block **104** includes a plurality of directional selector valves **16***a*, **16***d*, **16***e*, **16***g*, and **16***j* that are connected to the common hydraulic fluid supply line **307** and control the directions and the flow rates of hydraulic fluids to be supplied to the plurality of actuators **3***a*, **3***b*, **3***d*, **3***e*, and **3***g*, a plurality of directional selector valves **16***b*, **16***c*, **16***f*, **16***h* and **16***i* that are connected to the common hydraulic fluid supply line **308** and control the directions and the flow rates of hydraulic fluids to be supplied to the plurality of actuators **3***a*, **3***b*, **3***c*, **3***f*, and **3***h*, check valves **80***a*, **80***b*, **80***c*, **80***d*, **80***e*, **80***f*, **80***g*, **80***h*, **80***i*, and **80***j*, and a main relief valve **14** that is provided on the downstream of the common hydraulic fluid supply lines **307** and **308** through the check valves **80***e* and **80***f* and controls the pressure of the common hydraulic fluid supply lines **307** and **308** so as not to become equal to or higher than a set pressure.

To the downstream of the common hydraulic fluid supply line **307**, the directional selector valve **16***g* is connected on the most upstream, and the directional selector valves **16***a* and **16***d* are connected in parallel to each other to the downstream of the directional selector valve **16***g* through the check valves **80***a* and **80***b*, respectively. Further, the directional selector valves **16***j* and **16***e* are connected in tandem connection to the downstream of the directional selector valves **16***a* and **16***d* through the check valves **80***c* and **80***d* such that the directional selector valve **16***j* is positioned on the upstream side with respect to the directional selector valve **16***e*.

To the downstream of the common hydraulic fluid supply line **308**, the directional selector valves **16***c*, **16***b*, **16***i*, and **16***h* are connected in parallel to each other through the check valves **80***g*, **80***h*, **80***i*, and **80***j*, respectively, and the directional selector valve **16***f* is connected in tandem connection to the downstream of the directional selector valve **16***h*.

The plurality of the directional selector valves **16***a*, **16***d*, **16***e*, **16***g*, and **16***j* and the plurality of directional selector valves **16***b*, **16***c*, **16***f*, **16***h* and **16***i* are of the open center type, and the main pumps **122** and **222** and main pumps **422** and **622** and the control valve block **104** configure a hydraulic drive system of an open circuit.

The main pumps **122**, **222**, **422**, and **622** of the variable displacement type include regulator pistons **122***a*, **222***a*, **422***a*, and **622***a* for reducing, when the delivery pressure thereof becomes high, the displacement thereof to limit the torque, respectively.

Further, the main pump **122** and the main pump **422** include regulator pistons **122***b* and **422***b*, respectively, for reducing, if the delivery pressure of one of them becomes high, the displacement of the main pump of the other of them to limit the torque, and the main pump **222** and the main pump **622** include regulator pistons **222***b* and **622***b* for restricting, if the delivery pressure of one of them becomes high, the displacement of the main pump of the other of them to limit the torque.

The main pumps **122**, **222**, **422**, and **622** have springs **122***f*, **222***f*, **422***f*, and **622***f* for setting a limitation value q**1***limit, a limitation value q**2***limit, a limitation value q**3***limit, and a limitation value q**4***limit for the absorption horsepower, respectively.

Furthermore, the main pumps **122** and **422** include regulator pistons **122***c* and **422***c* for reducing the displacement of them by an external pressure (output pressure of a proportional solenoid valve **20** hereinafter described) to reduce the torque, and the main pumps **222** and **622** include regulator pistons **222***c* and **622***c* for reducing the displacement of them by an external pressure (output pressure of a proportional solenoid valve **21** hereinafter described) to reduce the torque.

Further, the hydraulic drive system of the present embodiment includes a hydraulic fluid supply line **31** to which hydraulic fluid delivered from the pilot pump **30** of the fixed displacement type is introduced, a gate lock valve **100** that is connected to the hydraulic fluid supply line **31** and performs selection in regard to whether the pilot hydraulic fluid line on the downstream side is to be connected to the hydraulic fluid supply line **31** or the tank, and a gate lock lever **24** disposed on the operator's seat entrance side of the hydraulic work machine for performing a selection operation of the gate lock valve **100**.

The pilot hydraulic fluid line on the downstream side of the gate lock valve **100** is connected to a plurality of pilot valves **60***a*, **60***b*, **60***c*, **60***d*, **60***e*, **60***f*, **60***g*, and **60***h* provided on a plurality of operation devices **522**, **523**, **524***a*, **524***b*, . . . (refer to FIG. **2**), and the plurality of pilot valves **60***a*, **60***b*, **60***c*, **60***d*, **60***e*, **60***f*, **60***g*, and **60***h* generate a pilot operation pressure according to an operation amount based on hydraulic fluid of the pilot hydraulic fluid line on the downstream side of the gate lock valve **100**, and the plurality of directional selector valves **16***a*, **16***b*, **16***c*, **16***d*, **16***e*, **16***f*, **16***g*, **16***h*, **16***i*, and **16***j* are controlled for selection by the pilot operation pressure.

The hydraulic drive system of the present embodiment further includes inverters **160**, **260**, **460**, and **660** for controlling the rotational speeds of the electric motors **101**, **201**, **401**, and **601**, a step-up/step-down chopper **161** for supplying electric power with a fixed voltage to the inverters **160** and **460**, another step-up/step-down chopper **261** for supplying electric power with a fixed voltage to the inverters **260** and **660**, an inverter **360** for controlling the rotational speed of the electric motor **301**, a step-up/step-down chopper **361** for supplying electric power with a fixed voltage to the inverter **360**, a power storage device **170** (first power storage device) connected so as to supply electric power to the electric motors **101** and **401** (first electric motors) through the step-up/step-down choppers **161** and the inverters **160** and **460**, a power storage device **270** (second power storage device) connected so as to supply electric power to the electric motors **201** and **601** (second electric motors) through the step-up/step-down chopper **261** and the inverters **260** and **660** and connected so as to supply electric power to the electric motor **301** through the step-up/step-down chopper **361** and the inverter **360**, battery management systems (BMS) **171** and **271** that output information such as a voltage, a temperature, and so forth of the power storage devices **170** and **270** to a controller **150** hereinafter described, a reference rotational speed instruction dial **56** for instructing a maximum speed of the plurality of actuators **3***a* to **3***h*, shuttle valves **19***a*, **19***b*, **19***c*, **19***d*, **19***e*, **19***f*, **19***g*, and **19***h* that select and output the highest pressure from among output pressures of the pilot valves **60***a*, **60***b*, **60***d*, **60***e*, and **60***g*, a pressure sensor **43** that detects the selected highest pressure, shuttle valves **19***i*, **19***j*, **19***k*, **191**, **19***m*, **19***n*, **19***o*, and **19***p* that select and output the highest pressure from among output pressures of the pilot valves **60***a*, **60***b*, **60***c*, **60***h*, and **60***f*, a pressure sensor **44** that detects the selected highest pressure, a pressure sensor **42** that is provided in the hydraulic fluid supply line **31** of the pilot pump **30** and detects a pressure of the hydraulic fluid supply line **31**, namely, a delivery pressure (hereinafter referred to suitably as pump pressure) of the pilot pump **30**, a proportional solenoid valve **20** that is connected to the pilot hydraulic fluid line on the downstream side of the gate lock valve **100** and decreases the pressure of the hydraulic fluid supply line **31**, to which hydraulic fluid is delivered from the pilot pump **30** of the fixed displacement type, to generate a torque controlling pressure to be introduced to the regulator pistons **122***c* and **422***c*, a proportional solenoid valve **21** that decreases the pressure of the hydraulic fluid supply line **31** to generate a torque controlling pressure to be introduced to the regulator pitons **222***c* and **622***c*, a controller **150** that receives signals from the battery management systems **171** and **271**, reference rotational speed instruction dial **56**, and pressure sensors **43** and **44** as inputs thereto to generate rotational speed instruction signals for the inverters **160**, **260**, **460**, and **660** and control signals for the proportional solenoid valves **20** and **21** from the input information to control the inverters **160**, **260**, **460**, and **660** and the proportional solenoid valves **20** and **21**, and another controller **55** that generates a rotational speed instruction signal for the inverter **360** on the basis of a detection signal of the pressure sensor **42** to control the inverter **360**.

FIG. **14** depicts a functional block diagram of the controller **150** in the fourth embodiment.

The controller **150** includes a proportional solenoid valve target current calculation section **151**, gain tables **152***a* and **152***b*, multiplication sections **153***a* and **153***b*, a rotational speed instruction value table **154**, and command value tables **155***a* and **155***b*.

Outputs of the battery management systems **171** and **271** are inputted to the proportional solenoid valve target current calculation section **151**, and an output of the proportional solenoid valve target current calculation section **151** is outputted to the proportional solenoid valves **20** and **21**.

Meanwhile, a maximum operation pilot pressure of the boom cylinder **3***a*, arm cylinder **3***b*, bucket cylinder **3***d*, swing cylinder **3***e* and right travel motor **3***g* is inputted to the controller **150** through the pressure sensor **43**, and a maximum operation pilot pressure of the boom cylinder **3***a* (in the raising direction only), arm cylinder **3***b*, swing motor **3***c*, blade cylinder **3***h* and left travel motor **3***f* is inputted to the controller **150** through the pressure sensor **44**.

Outputs from the pressure sensors **43** and **44** are inputted to the gain tables **152***a* and **152***b*, by which they are converted into gains N_gain**1** and N_gain**2** within a range of 0 to 100%, respectively. Further, an output from the reference rotational speed instruction dial **56** is converted into a reference rotational speed N_**0** by the rotational speed instruction value table **154**.

The reference rotational speed N_**0** that is an output of the rotational speed instruction value table **154** is multiplied by the gains N_gain**1** and N_gain**2** by the multiplication sections **153***a* and **153***b* to obtain target rotational speeds N**1***d* and N**2***d* for the inverters **160** and **260** and the inverters **460** and **660**, respectively.

The target rotational speeds N**1***d* and N**2***d* are inputted to the command value tables **155***a* and **155***b* and converted into command values Vinv**1** and Vinv**2** for the inverters **160** and **260** and the inverters **460** and **660**, respectively, and they are outputted to the inverters **160** and **260** and the inverters **460** and **660**.

FIG. **15** depicts a functional block diagram of the proportional solenoid valve target current calculation section **151** in the controller **150** in the fourth embodiment.

The proportional solenoid valve target current calculation section **151** includes first and second SOC estimation sections **151***a* and **151***b*, a differentiation section **151***c*, and first and second tables **151***d* and **151***e*.

Information (a voltage, a temperature and so forth) from the battery management systems **171** and **271** is inputted to the first and second SOC estimation sections **151***a* and **151***b* indicative of charge states of the power storage devices **170** and **270**, and a SOC**1** and a SOC**2** are outputted as SOCs of the first and second SOC estimation sections **151***a* and **151***b*, respectively. The SOC**1** and the SOC**2** are differentiated by the differentiation section **151***c* to obtain ΔSOC (=SOC**1**−SOC**2**). The difference ΔSOC is inputted to the first and second tables **151***d* and **151***e*, by which they are converted into target currents I**1** and I**2** (horsepower control amounts), which are outputted to the proportional solenoid valves **20** and **21**.

FIGS. **16**A and **16**B depict characteristics of the first and second tables **151***d* and **151***e*.

As depicted in FIG. **16**A, the characteristic of the first table **151***d* is set such that, when ΔSOC has a positive value, 0 is outputted as the target current I**1** of the proportional solenoid valve **20**, and when ΔSOC has a negative value, as ΔSOC decreases from the dead zone −ΔSOC_**2**, the target current I**1** of the proportional solenoid valve **20** increases, and when ΔSOC reaches −ΔSOC_**0** in the proximity of the maximum value −ΔSOC_max, a maximum value I_max is outputted as the target current I**1**.

As depicted in FIG. **16**B, the characteristic of the second table **151***e* is set reverse to the characteristic of the first table **151***d*. In particular, the characteristic of the second table **151***e* is set such that, when ΔSOC has a negative value, 0 is outputted as the target current I**2** of the proportional solenoid valve **21**, and when ΔSOC has a positive value, as ΔSOC increases from the dead zone ΔSOC_**2**, the target current I**2** of the proportional solenoid valve **21** increases, and when ΔSOC reaches ΔSOC_**0** in the proximity of the maximum value ΔSOC_max, a maximum value I_max is outputted as the target current I**2**.

FIG. **17** depicts a horsepower control characteristic by the regulator pistons **122***a* and **422***a*, regulator pistons **122***b* and **422***b*, and regulator pistons **122***c* and **422***c* of the main pumps **122** and **422** of the variable displacement type. Further, in FIG. **17**, a horsepower control characteristic by the regulator pistons **222***a* and **622***a*, regulator pistons **222***b* and **622***b*, and regulator pistons **222***c* and **622***c* of the main pumps **222** and **622** of the variable displacement type is depicted in parentheses.

The axis of abscissa of FIG. **17** depicts an average value Pa (=P**1**+P**2**/2) of delivery pressures P**1** and P**2** of the main pumps **122** and **422**, and a characteristic indicated by a solid line A**2** is a characteristic when the target current I**1** of the proportional solenoid valve **20** is 0 and the output pressure of the proportional solenoid valve **20** is 0. The regulator pistons **122***a*, **422***a*, **122***b*, and **422***b* have a characteristic that, after the average value Pa of the delivery pressures P**1** and P**2** exceeds Pa_a, as it becomes higher, the displacement qq**1** of the main pumps **122** and **422** decreases.

Further, as the target current I**1** of the proportional solenoid valve **20** increases and the output pressure of the proportional solenoid valve **20** becomes higher, the horsepower control characteristic becomes such a characteristic that it varies as indicated with a broken line B**2** or C**2** as indicated by an arrow mark in FIG. **17** to increase the degree of the limitation stronger and the displacement qq**1** of the main pumps **122** and **422** decreases.

This similarly applies also to the horsepower control characteristics by the regulator pistons **222***a* and **622***a*, regulator pistons **222***b* and **622***b*, and regulator pistons **222***c* and **622***c* of the main pumps **222** and **622** of the variable displacement type indicated in parentheses in FIG. **17**.

~Correspondence to Claims~

In the foregoing, the main pumps **122** and **222** (or **422** and **622**) of the variable displacement type are first and second hydraulic pumps, and hydraulic fluids delivered from the first and second hydraulic pumps are merged in the common hydraulic fluid supply line **307** (or **308**) and supplied to the plurality of flow control valves **16***a*, **16***b*, **16***d*, **16***e*, and **16***g* (or **16***a*, **16***b*, **16***c*, **16***f*, and **16***h*) and further supplied to the plurality of actuators **3***a*, **3***b*, **3***d*, **3***e*, **3***f*, and **3***g* (or **3***a*, **3***b*, **3***d*, **3***e*, and **3***g*).

The electric motors **101** and **201** (or **401** and **601**) are first and second electric motors that drive the main pumps **122** and **222** (or **422** and **622**) (first and second hydraulic pumps), respectively, and the power storage devices **170** and **270** are first and second power storage devices that supply electric power to the electric motors **101** and **201** (or **401** and **601**) (first and second motors), respectively.

The regulator piston **122***a* (or **422***a*) provides a first horsepower control device configured to decrease, when the delivery pressure of the main pump **122** (or **422**) (first hydraulic pump) increases, the delivery flow rate of the main pump **122** (or **422**) (first hydraulic pump) to control the absorption horsepower of the main pump **122** (or **422**) (first hydraulic pump) so as not to exceed the limitation value q**1***limit (first allowable value), and the regulator piston **222***a* (or **622***a*) provides a second horsepower control device configured to decrease, when the delivery pressure of the main pump **222** (or **622**) (second hydraulic pump) increases, the delivery flow rate of the main pump **222** (or **622**) (second hydraulic pump) to control the absorption horsepower of the main pump **222** (or **622**) (second hydraulic pump) so as not to exceed the limitation value q**2***limit (second allowable value).

The proportional solenoid valve target current calculation section **151** cooperates with the battery management systems **171** and **271** to provide a horsepower distribution control section configured to change at least one of the limitation value q**1***limit (first allowable value) and the limitation value q**2***limit (second allowable value) of the first and second horsepower control devices such that the charge state of the power storage device **170** (first power storage device) and the charge state of the power storage device **270** (second power storage device) become equal to each other.

Further, in the present embodiment, the main pumps **122** and **222** (or **422** and **622**) are hydraulic pumps of the variable displacement type, and the first horsepower control device are configured to control the displacements of the main pump **122** (or **422**) (first hydraulic pump) and the main pump **222** (or **622**) (second hydraulic pump) to control the absorption horsepower of the main pumps **122** and **222** (or **422** and **622**) (first and second hydraulic pumps) such that the absorption horsepower of the main pump **122** (or **422**) (first hydraulic pump) does not exceed the limitation value q**1***limit (first allowable value) and the absorption horsepower of the main pump **222** (or **622**) (second hydraulic pump) does not exceed the limitation value q**2***limit (second allowable value).

The gain table **152***a* (**152***b*), multiplication section **153***a* (**153***b*), rotational speed instruction value table **154**, command value table **155***a* (**155***b*), and inverter **160** (or **460**) provide a first flow control section configured to perform when at least one of the operation devices **522**, **523**, **524***a*, **524***b*, . . . is operated, positive flow control for controlling the delivery flow rate of the main pump **122** (or **422**) (first hydraulic pump) as the required flow rate by the operation device increases, and the gain table **152***a* (**152***b*), multiplication section **153***a* (**153***b*), rotational speed instruction value table **154**, command value table **155***a* (**155***b*), and inverter **260** (or **660**) provide a second flow control section configured to perform when at least one of the plurality of operation devices **522**, **523**, **524***a*, **524***b*, . . . is operated, positive flow control for controlling the delivery flow rate of the main pump **222** (or **622**) (second hydraulic pump) as the required flow rate by the operation device increases.

The first and second flow control sections described above are configured to control the rotational speeds of the main pumps **122** and **222** (or **422** and **622**) (first and second hydraulic pumps) respectively to control the delivery flow rates of the main pumps **122** and **222** (or **422** and **622**) (first and second hydraulic pumps) as the required flow rate increases, and the first and second horsepower control devices are configured to control the delivery flow rates of the main pumps **122** and **222** (or **422** and **622**) (first and second hydraulic pumps) respectively, which are controlled by positive flow control, such that the absorption horsepower of the main pump **122** (or **422**) (first hydraulic pump) does not exceed the limitation value q**1***limit (or q**3***limit) (first allowable value) and the absorption horsepower of the main pump **222** (or **622**) (second hydraulic pump) does not exceed the limitation value q**2***limit (or q**4***limit).

The battery management system **171** and the first SOC estimation section **151***a* in the proportional solenoid valve target current calculation section **151** of the controller **150** provide a first charge state estimation section configured to estimate the storage state of the power storage device **170** (first power storage device), and the battery management system **271** and the second SOC estimation section **151***b* in the proportional solenoid valve target current calculation section **151** of the controller **150** provide a second charge state estimation section configured to estimate the charge state of the power storage device **270** (second power storage device).

The differentiation section **151***c* in the proportional solenoid valve target current calculation section **151** of the controller **150** and the first and second tables **151***d* and **151***e* provide a horsepower control amount calculation section configured to calculate, when the charge state of the power storage device **170** (first power storage device) estimated by the first charge state estimation section is lower than the charge state of the power storage device **270** (second power storage device) estimated by the second charge state estimation section, a target current I**1** (first horsepower control amount) of the proportional solenoid valve **20** for decreasing the absorption horsepower of the main pump **122** (or **422**) (first hydraulic pump), and calculate, when the charge state of the power storage device **270** (second power storage device) estimated by the second charge state estimation section is lower than the charge state of the power storage device **170** (first power storage device) estimated by the first charge state estimation section, a target current I**2** (second horsepower control amount) of the proportional solenoid valve **21** for decreasing the absorption horsepower of the main pump **222** (or **622**) (second hydraulic pump).

The proportional solenoid valve **20** and the regulator piston **122***c* (or **422***c*) of the main pump **122** (or **422**) provide a first allowable value changing section configured to change the limitation value q**1***limit (or q**3***limit) (first allowable value) of the first horsepower control device on the basis of the target current I**1** (first horsepower control amount) of the proportional solenoid valve **20** calculated by the horsepower control amount calculation section. The proportional solenoid valve **21** and the regulator piston **222***c* (or **622***c*) of the main pump **222** (or **622**) provide a second allowable value changing section configured to change the limitation value q**2***limit (or q**4***limit) (second allowable value) of the second horsepower control device on the basis of the target current I**2** (second horsepower control amount) of the proportional solenoid valve **21** calculated by the horsepower control amount calculation section.

~Operation~

Operation of the fourth embodiment is described with reference to FIGS. **12** to **17**.

Similarly as in the third embodiment, the pressure of the hydraulic fluid supply line **31** is kept at the pilot primary pressure Ppi**0** by controlling the rotational speed of the electric motor **301** for driving the pilot pump **30** of the fixed displacement type.

(a) Case where the SOCs of the Power Storage Device **170** and the Power Storage Device **270** are Equal to Each Other

FIG. **15** depicts a functional block diagram of the proportional solenoid valve target current calculation section **151**.

First, in the case where the charge state SOC**1** of the power storage device **170** estimated by the first SOC estimation section **151***a* and the charge state SOC**2** of the power storage device **270** estimated by the second SOC estimation section **151***b* are equal to each other, since SOC**1**=SOC**2** is satisfied, ΔSOC computed by the differentiation section **151***c* becomes ΔSOC=SOC**1**−SOC**2**=0, and since the current commands to the proportional solenoid valves **20** and **21** become 0 from the characteristics of the tables **151***d* and **151***e* depicted in FIG. **16**, the displacement of the main pumps **122** and **422** becomes qqmax1 and the displacement of the main pumps **222** and **622** become qqmax2 as depicted in FIG. **17**.

<(a-1) Case where all Operation Levers are Neutral>

As described hereinabove, since a maximum operation pilot pressure of the boom cylinder **3***a*, arm cylinder **3***b*, bucket cylinder **3***d*, swing cylinder **3***e*, and right travel motor **3***g* is inputted to the controller **150** through the pressure sensor **43**, and a maximum operation pilot pressure of the boom cylinder **3***a* (only in the raising direction), arm cylinder **3***b*, swing motor **3***c*, blade cylinder **3***h*, and left travel motor **3***f* is inputted to the controller **150** through the pressure sensor **44**, in the case where all operation levers are neutral, the gains N_gain**1** and N_gain**2** are kept at a minimum value (for example, 0%) by the gain tables **152***a* and **152***b* in the controller **150**.

Further, an output from the reference rotational speed instruction dial **56** is converted into a reference rotational speed N_**0** by the rotational speed instruction value table **154** and multiplied by the N_gain**1** and N_gain**2** by the multiplication sections **153***a* and **153***b* so as to be converted into target rotational speeds N**1***d* and N**2***d* and is further converted into an output Vinv**1** to the inverters **160** and **260** and an output Vinv**2** to the inverters **460** and **660** by the command value tables **155***a* and **155***b*. Therefore, when all operation levers are neutral, since N_gain**1** and N_gain**2** are kept at a minimum value (for example, 0%) as described hereinabove, both of the output Vinv**1** to the inverters **160** and **260** and the output Vinv**2** to the inverters **460** and **660** are kept at a minimum value.

Since Vinv**1** and Vinv**2** are kept at their minimum value, all of the electric motors **101**, **201**, **401**, and **601** drive the main pumps **122**, **222**, **422**, and **622** with a minimum rotational speed to supply hydraulic fluids.

Since the main pumps **122**, **222**, **422** and **622** are driven with a minimum rotational speed, also the flow rates of hydraulic fluids delivered from the main pumps are minimized.

Further, hydraulic fluids delivered from the main pumps **122** and **222** pass the hydraulic fluid supply lines **105***a* and **105***b*, respectively, and are merged in the hydraulic fluid supply line **307** and then supplied to the P**1** port of the control valve block **104**.

Since all operation levers are neutral, the hydraulic fluid supplied to the P**1** port is discharged to the tank through the neutral circuits of the directional selector valves **16***g*, **16***d*, **16***a*, **16***j*, and **16***e*.

On the other hand, hydraulic fluids delivered from the main pumps **422** and **622** pass through the hydraulic fluid supply lines **205***a* and **205***b*, respectively, and are merged in the hydraulic fluid supply line **308** and then supplied to the P**2** port of the control valve block **104**.

Since all operation levers are neutral, the hydraulic fluid supplied to the P**2** port is discharged to the tank through the neutral circuits in the directional selector valves **16***c*, **16***b*, **16***i*, **16***h*, and **16***f*.

<(a-2) Case where Boom Raising is Operated>

A case is considered in which the operation lever for the boom is inputted in a raising direction, namely, in a direction in which the boom cylinder **3***a* extends.

The pilot valve **60***a* for the boom operation is operated, and an operation pilot pressure is generated in a**1**.

Since the operation pilot pressure a1 is applied, both of the directional selector valves **16***a* and **16***i* for the boom select the rightward direction in the figure, and hydraulic fluids supplied from the main pumps **122** and **222** of the variable displacement type through the hydraulic fluid supply line **307** and hydraulic fluids supplied from the main pumps **422** and **622** of the variable displacement type through the hydraulic fluid supply line **308** are merged and supplied to the bottom side of the boom cylinder **3***a*.

On the other hand, the output pressure of the pilot valve **60***a* for the boom operation is introduced to the pressure sensor **43** through the shuttle valves **19***a*, **19***e*, **19***f*, **19***g*, and **19***h* and simultaneously introduced to the pressure sensor **44** through the shuttle valve **19***p*.

Output Vpc**1** and Vpc**2** of the pressure sensors **43** and **44** are introduced to the gain tables **152***a* and **152***b*, from which N_gain**1** and N_gain**2** are outputted in response to an output of the pilot valve **60***a* for the boom operation.

For example, in the case where the boom raising operation is a full operation, both of N_gain**1** and N_gain**2** become a maximum value 100%.

On the other hand, an output from the reference rotational speed instruction dial **56** to be converted into a reference rotational speed N_**0** by the rotational speed instruction value table **154** is multiplied by the N_gain**1** and N_gain**2** described above by the multiplication sections **153***a* and **153***b* so as to be converted into target rotational speeds N**1***d* and N**2***d*, respectively, and are further converted into an output Vinv**1** to the inverters **160** and **260** and an output Vinv**2** to the inverters **460** and **660** by the command value tables **155***a* and **155***b*, respectively. Therefore, in the case where the boom raising operation is a full operation, since N_gain **1** and N_gain**2** are kept at a maximum value (for example, 100%), both of the input Vinv**1** to the inverters **160** and **260** and the output Vinv**2** to the inverters **460** and **660** come to have a value same as that of an input from the reference rotational speed instruction dial **56**.

Further, in the case where the boom raising operation is a half operation and N_gain**1** and N_gain**2** outputted from the gain tables **152***a* and **152***b* are 50%, since the target rotational speeds N**1***d* and N**2***d* become values obtained by multiplying the reference rotational speed N_**0** from the reference rotational speed instruction dial **56** by 50%, N**1***d*=N**2***d*=N_**0**/2 is satisfied, and the electric motors **101**, **201**, **401**, and **601** individually rotate at a rotational speed equal to one half the reference rotational speed N_**0**.

In the case where the boom raising operation is performed in this manner, as the output of the pilot valve **60***a* for the boom operation increases, the rotational speed of the electric motors **101**, **201**, **401**, and **601** increases, and the extension speed of the boom cylinder **3***a* can be adjusted by the magnitude of the operation lever input.

Further, if it is assumed that the load pressure of the boom cylinder **3***a* upon boom raising is Pa_bmr, then upon a boom raising operation, both of the delivery pressures of the main pumps **122** and **222** and of the main pumps **422** and **622** become equal to the load pressure Pa_bmr of the boom cylinder **3***a*.

Here, in the case where the target current command values I1 and I2 of the proportional solenoid valves **20** and **21** are in the minimum, for example, 0, the horsepower control characteristic of the main pumps **122**, **222**, **422**, and **622** of the variable displacement type becomes such as indicated by a solid line A**2** in FIG. **17**, and at the pump delivery pressure of Pa_bmr, the displacement of the main pumps **122** and **222** of the variable displacement type is limited to qqbmr1 and the displacement of the main pumps **422** and **622** of the variable displacement type is limited to qqbmr2.

In this manner, when the boom raising operation is performed, the main pumps **122**, **222**, **422**, and **622** supply hydraulic fluids with displacements that depend upon respective horsepower control characteristics at a rotational speed according to a boom raising operation input.

(b) Case where the SOC of the Power Storage Device **170** is Greater than that of the Power Storage Device **270**

A case is considered in which, the relationship between the charge state SOC**1** of the power storage device **170** estimated by the first SOC estimation section **151***a* and the charge state SOC**2** of the power storage device **270** estimated by the second SOC estimation section **151***b* is SOC1>SOC2.

Since SOC**1**>SOC**2**, ΔSOC computed by the differentiation section **151***c* becomes ΔSOC=SOC**1**−SOC**2**>0, namely, has a positive value, the current command value I**1** for the proportional solenoid valve **20** becomes the minimum I1=0 and the current command value I**2** for the proportional solenoid valve **21** has a value according to the characteristic of the table **151***e* depicted in FIG. **16**. In the case ΔSOC>ΔSOC_0, the current command value I2 for the proportional solenoid valve **21** becomes I2=I_max.

Since the current command value I**1** for the proportional solenoid valve **20** is 0, the main pumps **122** and **422** of the variable displacement type have a characteristic indicated by a solid line A2 of a) I1=0 in FIG. **17**.

On the other hand, since the current command value I2 of the proportional solenoid valve **21** is I_max, the main pumps **222** and **622** of the variable displacement type have a characteristic indicated by a broken line C2 of c) I2=I_max in FIG. **17**.

<(b-1) Case where all Operation Levers are Neutral>

In the case where all operation levers are neutral, similarly as in the case of (a) SOC1=SOC2 described above, all of the main pumps **122**, **222**, **422**, and **622** of the variable displacement type deliver a minimum flow rate at a minimum rotational speed.

<(b-2) Case where Boom Raising is Operated>

Basic operation where the rotational speeds of the electric motors **101**, **201**, **401**, and **601** for driving the main pumps **122**, **222**, **422**, and **622** of the variable displacement type increases in response to an operation lever input amount for boom raising is similar to that in the case of (a) SOC1=SOC2 described hereinabove.

In the case where SOC1>SOC2 and ΔSOC (=SOC1−SOC2) is ΔSOC_**0** as described above, the current command value I1 for the proportional solenoid valve **20** becomes the minimum value 0 and the main pumps **122** and **422** follow the characteristic of a) I1=0 in FIG. **17**. Meanwhile, since the current command value I2 for the proportional solenoid valve **21** becomes 12=I_max, the main pumps **222** and **622** follow the characteristic indicated by the broken line C**2** of c) I2=I_max.

If the pump load pressure upon boom raising operation is Pa_bmr, then the displacement of the main pumps **122** and **422** of the variable displacement type becomes qqbmr1 in FIG. **17**.

On the other hand, the displacement of the main pumps **222** and **622** of the variable displacement type becomes qqbmr2_min, and qqbmr2_min<qqbmr1 is satisfied.

Although the rotational speeds of the electric motors **101** and **401** for driving the main pumps **122** and **422** of the variable displacement type and the electric motors **201** and **601** for driving the main pumps **422** and **622** of the variable displacement type depends only on the boom raising operation amount, since the displacement qq**1** of the main pumps **122** and **422** of the variable displacement type and the displacement qq**2** of the main pumps **222** and **622** of the variable displacement type have a difference therebetween as described above, the flow rates of hydraulic fluids delivered by the main pumps **222** and **622** of the variable displacement type are lower than the flow rates of hydraulic fluids delivered by the main pumps **122** and **422** of the variable displacement type.

Although the power consumption of a pump increases in proportion to the pressure×the flow rate and, upon boom raising operation, the pressures of the hydraulic fluid supply lines **105** and **205** are equal to each other, since the delivery flow rates of the main pumps **222** and **622** are lower than the delivery flow rates of the main pumps **122** and **422** as described hereinabove, the power consumption of the main pumps **222** and **622** is lower than the power consumption of the main pumps **122** and **422**.

Therefore, the electric power consumption of the power storage device **270** from which electric power is supplied to the main pumps **222** and **622** is lower than the electric power consumption of the power storage device **170** from which electric power is supplied to the main pumps **122** and **422**.

Since the electric power consumption of the power storage device **270** from which electric power is supplied to the main pump **202** is lower than the electric power consumption of the power storage device **170** from which electric power is supplied to the main pump **102**, the rate at which the SOC**2** of the power storage device **270** decreases is lower than the rate at which the SOC**1** of the power storage device **170** decreases, and this continues until after the SOC**1** and the SOC**2** become equal to each other.

If SOC**1**=SOC**2** is satisfied, then operation same as that in the case of (a) is performed.

(c) Case where the SOC of the Power Storage Device **270** is Greater than that of the Power Storage Device **170**

Operation is substantially same as that of (b) except that, since SOC**1**>SOC**2** changes to SOC**1**<SOC**2**, the relationship between the main pumps **122** and **422** and the main pumps **222** and **622** is reversed.

~Advantage~

Also with the present embodiment, advantages similar to those obtained by the first embodiment are obtained over the conventional hydraulic drive system of an open circuit including two pumps.

In particular, since the hydraulic drive system is configured such that, in addition to the main pumps **122** and **422**, electric motors **101** and **401**, and power storage device **170**, the main pumps **222** and **622**, electric motors **401** and **601**, and power storage device **270** as well as the common hydraulic fluid supply line **307** in which hydraulic fluids delivered from the main pumps **122** and **222** are merged and the common hydraulic fluid supply line **308** in which hydraulic fluids delivered from the main pumps **422** and **622** are merged are provided and the merged hydraulic fluid is supplied to the plurality of directional selector valves (flow control valves) **16***a*, **16***d*, **16***e*, **16***g*, and **16***j* or the plurality of directional selector valves (flow control valves) **16***b*, **16***c*, **16***f*, **16***h*, and **16***i* and is further supplied to the plurality of actuators **3***a*, **3***b*, **3***c*, **3***d*, **3***e*, and **3***g* or the plurality of actuators **3***a*, **3***b*, **3***c*, **3***f*, and **3***h*, the rated voltage of various electric equipment such as power storage devices can be made common to that of an electrically-driven hydraulic work machine that is capable of being operated with lower horsepower.

Further, since the proportional solenoid valve target current calculation section **151** and the proportional solenoid valves **20** and **21** are provided and, in the case where the charge state of one of the power storage devices **170** and **270** becomes lower than the charge state of the other one of the power storage devices **170** and **270**, the limit value q**1***limit (q**2***limit) of horsepower control of the main pumps **122** and **222** (or **422** and **622**) is changed such that the charge state of the power storage device **170** and the charge state of the power storage device **270** become equal to each other thereby to suppress the power consumption of the hydraulic pump whose charge state is lower, even in the case where there is a difference in mechanical efficiency of the main pumps **122** and **222** or **422** and **622** and efficiency of electric equipment such as the inverters **160** and **260** or **460** and **660** or the step-up/step-down choppers **161** and **261**, or even in the case where there is a difference between the electric power consumption amounts or the charge state characteristics of the power storage devices **170** and **270**, the difference gradually decreases while the charge states of the power storage devices **170** and **270** are controlled so as to become equal to each other. Therefore, the power storage situation of one of the power storage devices **170** and **270** is prevented from being significantly degraded, and the period of time within which the actuators of the electrically-driven hydraulic work machine can obtain a predetermined speed can be extended.

It is to be noted that, also in the present embodiment, similarly as in the case of the first embodiment, only one of the first and second tables **151***d* and **151***e* (for example, the first table **151***d*) and a corresponding one of the proportional solenoid valves **20** and **21** (for example, the proportional solenoid valve **20**) may be provided such that, only in the case where the charge state of one (for example, the power storage device **270**) of the two power storage devices **170** and **270** becomes lower than the charge state of the other power storage device (for example, the power storage device **170**), the limit value q**1***limit (first allowable value) is changed such that the charge state of the power storage device **170** and the charge state of the power storage device **270**) become equal to each other. Also this makes it possible to prevent that the charge state of one (for example, the power storage device **270**) of the power storage devices **170** and **270** becomes significantly lower than the charge state of the other one (for example, the power storage device **170**), and the period of time within which the actuators of the electrically-driven hydraulic work machine can obtain a predetermined speed can be extended.

DESCRIPTION OF REFERENCE CHARACTERS

3*a* to **3***h*: Actuator
4: Control valve block
6*a* to **6***h*: Flow control valve
7*a* to **7***h*: Pressure compensating valve
9*a* to **9***g*: Shuttle valve
14, **32**: Relief valve

16a to **16**j: Directional selector valve (flow control valve)
19a to **19**p: Shuttle valve
20, **21**: Proportional solenoid valve (horsepower distribution control section)
24: Gate lock lever
31: Hydraulic fluid supply line (pilot)
40, **41**, **42**: Pressure sensor
43, **44**: Pressure sensor
50, **55**: Controller
51: Virtual limitation torque calculation section (horsepower distribution control section)
51a: First SOC estimation section (first charge state estimation section)
51b: Second SOC estimation section (second charge state estimation section)
51c: Differentiation section (horsepower control amount calculation section)
51d: First table (horsepower control amount calculation section)
51e: Second table (horsepower control amount calculation section)
52, **53**: Electric motor rotational speed control section
52r, **53**r: Variable horsepower control table (first and second horsepower control devices; horsepower control section; first and second allowable value changing section)
52s, **53**s: Minimum value selection section (first and second horsepower control devices)
52a to **52**m: Table and other calculation factors (first flow control section)
53a to **53**m: Table and other calculation factors (second flow control section)
56: Reference rotational speed instruction dial
80a to **80**j: Check valve
100: Selector valve
101, **201**: Electric motor (first and second electric motors)
102, **202**: Main pump of the fixed displacement type (first and second hydraulic pumps)
104: Control valve block
105, **205**: Hydraulic fluid supply line (main)
122, **422**: Main pump of the variable displacement type (first hydraulic pumps)
122a, **222**a, **422**a, **622**a: Regulator piston (first and second horsepower control devices)
130, **230**, **30**: Pilot pump of the variable displacement type
133, **233**, **180**, **280**: Check valve
150: Controller
151: Proportional solenoid valve target current calculation section (horsepower distribution control section)
152a, **152**b to **155**a, **155**b: Table and multiplication section (first and second flow control sections)
160, **260**, **360**: Inverter
161, **261**, **361**: Step-up/step-down chopper
170, **270**: Power storage device (first and second power storage devices)
171, **271**: Battery management system (BMS)
180, **280**: Check valve (first and second check valves)
222, **622**: Main pump of the variable displacement type (second hydraulic pump)
301: Electric motor (third electric motor)
305, **307**, **308**: Common hydraulic fluid supply line
401, **601**: Electric motor (first and second electric motors)
460, **660**: Inverter

The invention claimed is:

1. A hydraulic drive system for an electrically-driven hydraulic work machine including a first hydraulic pump, a plurality of actuators driven by hydraulic fluid supplied from the first hydraulic pump, a plurality of flow control valves that control directions and flow rates of the hydraulic fluids to be supplied to the plurality of actuators, a first electric motor that drives the first hydraulic pump, a first power storage device that supplies electric power to the first electric motor, and a first horsepower control device configured to decrease, when a delivery pressure of the first hydraulic pump increases, a delivery flow rate of the first hydraulic pump to control an absorption horsepower of the first hydraulic pump so as not to exceed a first allowable value,

the hydraulic drive system comprising:

a second hydraulic pump;

a common hydraulic fluid supply line in which hydraulic fluids delivered from the first and second hydraulic pumps are merged and are supplied to the plurality of flow control valves;

a second electric motor that drives the second hydraulic pump;

a second power storage device that supplies electric power to the second electric motor;

a second horsepower control device configured to decrease, when a delivery pressure of the second hydraulic pump increases, a delivery flow rate of the second hydraulic pump to control an absorption horsepower of the second hydraulic pump so as not to exceed a second allowable value; and

a controller including a horsepower distribution control section configured to change at least one of the first and second allowable values of the first and second horsepower control devices such that a charge state of the first power storage device and a charge state of the second power storage device become equal to each other.

2. The hydraulic drive system for an electrically-driven hydraulic work machine according to claim **1**, wherein

each of the first and second hydraulic pumps is a hydraulic pump of fixed displacement type, and

the first and second horsepower control devices configured to control rotational speeds of the first and second hydraulic pump respectively to control the absorption horsepowers of the first and second hydraulic pumps such that the absorption horsepower of the first hydraulic pump does not exceed the first allowable value and the absorption horsepower of the second hydraulic pump does not exceed the second allowable value.

3. The hydraulic drive system for an electrically-driven hydraulic work machine according to claim **1**, wherein

each of the first and second hydraulic pumps is a hydraulic pump of variable displacement type, and

the first and second horsepower control devices are configured to control displacements of the first and second hydraulic pump respectively to control the absorption horsepowers of the first and second hydraulic pumps such that the absorption horsepower of the first hydraulic pump does not exceed the first allowable value and the absorption horsepower of the second hydraulic pump does not exceed the second allowable value.

4. The hydraulic drive system for an electrically-driven hydraulic work machine according to claim **1**, further comprising:

a plurality of operation devices that command operations of the plurality of actuators, wherein

the controller further includes

a first flow control section configured to perform, when at least one of the plurality of operation devices is operated, load sensing control for controlling a delivery flow rate of the first hydraulic pump such that the delivery pressure of the first hydraulic pump becomes higher by a target differential pressure than a highest load pressure of the plurality actuators, and

a second flow control section configured to perform, when at least one of the plurality of operation devices is operated, load sensing control for controlling a delivery flow rate of the second hydraulic pump such that the delivery pressure of the second hydraulic pump becomes higher by the target differential pressure than the highest load pressure of the plurality of actuators,

the first and second flow control sections are configured to control rotational speeds of the first and second hydraulic pumps respectively to control the delivery flow rates of the first and second hydraulic pumps such that the delivery pressures of the first and second hydraulic pumps become higher by the target differential pressure than the highest load pressure of the plurality of actuators, and

the first and second horsepower control devices are configured to control the delivery flow rates of the first and second hydraulic pumps respectively, which are controlled by the load sensing control, such that the absorption horsepower of the first hydraulic pump does not exceed the first allowable value and the absorption horsepower of the second hydraulic pump does not exceed the second allowable value.

5. The hydraulic drive system for an electrically-driven hydraulic work machine according to claim **1**, further comprising:

a plurality of operation devices that command operations of the plurality of actuators, wherein

the controller further includes

a first flow control section configured to perform, when at least one of the plurality of operation devices is operated, positive flow control for controlling a delivery flow rate of the first hydraulic pump as a required flow rate by the operation device increases, and

a second flow control section configured to perform, when at least one of the plurality of operation devices is operated, positive flow control for controlling a delivery flow rate of the second hydraulic pump as a required flow rate by the operation device increases,

the first and second flow control sections are configured to control rotational speeds of the first and second hydraulic pumps respectively to control the delivery flow rates of the first and second hydraulic pumps as the required flow rate increases, and

the first and second horsepower control devices are configured to control the delivery flow rates of the first and second hydraulic pumps respectively, which are controlled by the positive flow control, such that the absorption horsepower of the first hydraulic pump does not exceed the first allowable value and the absorption horsepower of the second hydraulic pump does not exceed the second allowable value.

6. The hydraulic drive system for an electrically-driven hydraulic work machine according to claim **1**, wherein

the horsepower distribution control section of the controller includes

a first charge state estimation section configured to estimate a charge state of the first power storage device,

a second charge state estimation section configured to estimate a charge state of the second power storage device,

a horsepower control amount calculation section configured to calculate, when the charge state of the first power storage device estimated by the first charge state estimation section is lower than the charge state of the second power storage device estimated by the second charge state estimation section, a first horsepower control amount for decreasing the absorption power of the first hydraulic pump and calculate, when the charge state of the second power storage device estimated by the second charge state estimation section is lower than the charge state of the first power storage device estimated by the first charge state estimation section, a second horsepower control amount for decreasing the absorption power of the second hydraulic pump,

a first allowable value changing section configured to change the first allowable value for the first horsepower control device on the basis of the first horsepower control amount calculated by the horsepower control amount calculation section, and

a second allowable value changing section configured to change the second allowable value for the second horsepower control device on the basis of the second horsepower control amount calculated by the horsepower control amount calculation section.

7. The hydraulic drive system for an electrically-driven hydraulic work machine according to claim **1**, further comprising:

a first hydraulic fluid supply line that introduces the hydraulic fluid from the first hydraulic pump to the common hydraulic fluid supply line;

a second hydraulic fluid supply line that introduces the hydraulic fluid from the second hydraulic pump to the common hydraulic fluid supply line;

a first check valve provided in the first hydraulic fluid supply line to block a flow of the hydraulic fluid from the common hydraulic fluid supply line to the first hydraulic pump; and

a second check valve provided in the second hydraulic fluid supply line for blocking a flow of the hydraulic fluid from the common hydraulic fluid supply line to the second hydraulic pump.

8. The hydraulic drive system for an electrically-driven hydraulic work machine according to claim **1**, further comprising:

a pilot pump of fixed displacement type that generates a pilot primary pressure as a primary pressure for a plurality of operation devices that command operation of the plurality of actuators;

a third electric motor to which electric power is supplied from one of the first and second power storage device to drive the pilot pump; and

a pilot pump control device including a controller configured to control a rotational speed of the third electric motor such that the pilot primary pressure generated by the pilot pump becomes equal to a target pressure.

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